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ISSUES OF THE JOURNAL WANTED

The Society wishes to secure copies of the following issues of The Journal: August 1911, October 1911, and January 1912. These will be purchased at 25 cents a copy. The January 1912 issue is especially desired.

SPRING MEETING IN CLEVELAND

The date for the Spring Meeting, previously announced for the first week in June, has been changed to the last week of May, opening on Tuesday, May 28, and closing on Friday, May 31. Every effort is being put forth by the local members in Cleveland and by the Committee on Meetings to make this one of the great conventions of the Society. The situation of Cleveland on the lake front, the variety and importance of its industrial works, and the attractiveness of the city as a whole all contribute to make it one of the most interesting cities in the country for a meeting of engineers. There is a large and enthusiastic body of engineers resident in the city to welcome the visiting members and their guests and to extend to them the hospitality for which the city is noted. The time at which the meeting is to be held is usually one of the most delightful in the whole year in this locality and everything points toward a most enjoyable occasion. The program committee have important arrangements in hand which they are at work upon but which are not far enough along to permit of announcement at the time of going to press. The plans for the reception of the visitors are having the personal

attention of Past-President Ambrose Swasey, Chairman of the executive committee in Cleveland, Robert H. Fernald, Vice-Chairman, F. W. Ballard, Secretary, and R. B. Sheridan, Treasurer; besides the coöperation of Worcester R. Warner and S. T. Wellman, Past-Presidents, and other members who have been assigned to the various committees. The subjects that will come up for discussion at the professional sessions include recent developments in steam turbines, both here and abroad; modern flour-milling processes; important advances in the analysis of steam engine performance; hydraulic governing; recent data on the thermal properties of steam; data on the temperature of condensing water reservoirs; discoveries and new processes in chilling cast iron; and numerous other topics upon which papers are in prospect or have already been submitted. In fact, the Committee on Meetings is somewhat embarrassed by the wealth of material and a selection of papers will probably have to be made, having in mind the diversity of interests in the Society at large. A more complete statement of the subjects for discussion and of the program of arrangements will be made in the May number of The Journal. The essential part of the present announcement is the opening statement with regard to the change of date; with the further recommendation that every member within reach of Cleveland who can possibly attend reserve the four days from May 28 to May 31 for this purpose. Papers cannot be considered for the meeting later than April 15.

APRIL MEETING IN NEW YORK

The Society is to have the honor of entertaining at the time of the New York monthly meeting, April 9, the Commission of the German Museum of Munich of masterpieces of natural science and technical sciences, consisting of the following gentlemen: Dr. O. von Miller, Imperial Counsellor and President des Vereines deutscher Ingenieure; Dr. W. von Borscht, Lord Mayor, Privy Court Counsellor; Dr. W. von Dyck, Privy Counsellor Professor; Dr. Rudolf Diesel, Doctor of Engineering, inventor of the Diesel Engine, Herr Gelius, Constructing Architect, and Herr Schirmann, Director of the Library of the Museum.

It is expected that addresses will be made by certain of the visiting members of the Commission. This will be one of the most important meetings of the year in New York and postal-card notices will be issued later containing the detailed announcement.

During the month of April, honorary membership will be conferred upon Dr. Rudolf Diesel at a meeting to be held in the Engineering Societies Building, at which Dr. Diesel will probably make an address upon the Diesel Engine.

MEETING IN NEW HAVEN

As elsewhere announced, the members of the Society in New Haven have arranged a meeting for Wednesday, April 17, partaking of the nature of a convention, with afternoon and evening sessions. These will be held in the Mason Laboratory of Mechanical Engineering of the Sheffield Scientific School and the general topic for discussion will be Manufacturing Costs. The afternoon sessions will be devoted to a discussion of the development of manufacturing in New England, covering the production of typical New England products, such as hardware, firearms, typewriters, machine tools, and brass goods. In the evening the question to be brought forward for discussion will be the extent to which scientific management is applicable to small industries, a phase of the subject upon which definite information is much in demand. Dinner will be served at six o'clock in the Yale Dining Club.

This plan of holding all-day meetings of the Society is a noteworthy step in the extension of the Society's influence through meetings in various cities. An incident of this kind which should not be overlooked, is the annual meeting held by the student section of the Society at the University of Kansas, previously reported in these pages, where an entire day was devoted to professional meetings, an inspection of mechanical equipment and a dinner.

REQUEST FROM NEW YORK PUBLIC LIBRARY

The New York Public Library wishes to supplement its files of technical periodicals and similar collection of "Source Material" with the monographs and reports on various engineering enterprises that appear in print from time to time. Publications of this sort coming through periodicals and the regular book trade are bought by the Library; but the separate pamphlet reports that represent so large a part of the literature of this kind are not handled by the regular book trade and can be secured only through the courtesy and thoughtfulness of the engineers making such reports.

As most of our members know, the New York Public Library is almost adjacent to the Library of the Engineering Societies, and the

most intimate coöperation prevails between the two institutions. It is therefore urged that all members broaden this friendly relation by contributing to the New York Public Library such reports as may be issued under their direction as well as to the Library in this building.

All documents of this kind are properly indexed and their preservation will be a lasting benefit to the donor as well as to the future student working along the same lines.

DISPLAY OF ERICSSON MODELS

On Saturday, March 9, the fiftieth anniversary of the battle of the Monitor and the Merrimac, a display of a series of models of inventions and scientific instruments made by Captain John Ericsson was opened to the public in the Engineering Societies Building. These models, which were made for the Centennial Exhibition in Philadelphia in 1876, subsequently came into the possession of Mr. George H. Robinson, and were presented by him in 1890 to the Metropolitan Museum of New York, which in 1907 gave them to the United Engineering Societies. They include a model of parts of the Monitor engine, showing the rockshaft, a globe actinometer, a solar calorimetor, a caloric engine, a galvanometer, a portable solar calorimeter, an aero-steam engine, a galvanometer cone, the rotary gun-carriage and transit platform of the Spanish gunboat Torpedo, made in 1873, the parallactic mechanism for measuring radiation of parts of a solar disk, a distance instrument for measuring distances at sea, made in 1841, and a dynamic register for measuring the relative power of currents of water and vapor, in addition to three cups, made respectively from the wood of the Congress, the Merrimac and the Cumberland, and presented to Captain Ericsson by the United States Government. A wooden model of the Monitor, the property of the Society, was also shown. This was presented to the Society in 1893 by Thomas F. Rowland, the builder of the Monitor at the Continental Iron Works.

John Ericsson became a member of the Society in 1882, and though offered honorary membership as well as other honors, he steadfastly refused to be so ranked, preferring to be considered simply as an active member. An interesting account of his life appears in Transactions Volume 10.

The Society also has in its possession a fine painting of Ericsson

which has an interesting history. In 1862 the artist Ballin of Stockholm, knowing Ericsson's aversion to sitting for a portrait, but being desirous of painting the great engineer, secured several interviews with him under the guise of a promoter of an alleged invention for an expansive bullet. During these interviews, he was able to fix the features of Ericsson so in his mind that he could afterwards reproduce them perfectly on the canvas. When the painting was completed he called upon Ericsson, carrying his masterpiece under his arm, only to be driven out into the street.

In 1895 the portrait was discovered by Professor F. R. Hutton, Secretary of the Society, in the rooms of a second-hand dealer, and was purchased and presented to the Society. The artist afterward called at the rooms of the Society and positively identified the picture. In 1865 the picture was presented to the Sanitary Fair held in Brooklyn, the proceeds of which were devoted to the hospital wards, and was purchased by Aaron Claflin, and later sold at auction in the Willoughby Street Auction Rooms, Brooklyn. It is possible that this is the portrait which was purchased by New York citizens and subsequently presented to Captain Ericsson. The painting, which is bust length, shows clearly Ericsson's strong and masterful features, with a glimpse also of those qualities as a dreamer which made him a great inventor.

The Society owns also a fine bust of Ericsson, made by John Kneeland, and presented by James M. Dodge, Past-President.

FOREIGN REVIEW

Attention is called to the new section of The Journal, The Foreign Review, which has appeared in the issues of this year.

The object of the Review is to show what the engineering world is doing in the non-English speaking countries, and it is made possible through the large number of foreign periodicals received in the Library of the Engineering Societies, numbering some two-hundred and twenty. These periodicals are gone over day by day as received and brief abstracts of the leading articles made in English, giving, where possible, engineering data, formulæ, tables, etc. Reference is made to publications in the English language, if such exist, containing fuller information on the subject treated. Measures are given both in original units and English equivalents and, wherever practicable, drawings are reproduced from the original articles.

The Review has necessarily been started in a small way, but the

Publication Committee hope to increase its scope and usefulness as the income of the Society permits. Inasmuch as but few engineers have these foreign periodicals at their disposal without a visit to a library, it is believed that the publication of this matter in The Journal will be welcomed by the membership with the ultimate result of extending very materially the usefulness of the Library of the Engineering Societies to those residing in different parts of the country. Suggestions are solicited for making the Review better and more useful. In connection with this work, also, the library offers to supply complete translations of foreign articles at moderate cost.

VISITS BY THE SECRETARY

Supplementing the tour of student branches made by the Secretary the early part of the year, a visit was made on March 13 to the Rensselaer Polytechnic Institute, where the Secretary spoke to the membership on the advantages of student engineering societies.

On March 15 a meeting of the Student Branch at Cornell University was held in the afternoon, and in the evening the annual Sibley Banquet, both of which were addressed by the Secretary. Cornell has the largest branch affiliated with the Society and its membership is enthusiastic and wide-awake. Six hundred guests attended the banquet, which was also addressed by Prof. C. F. Hirshfeld of the department of mechanical engineering, Prof. C. L. Durham of the department of arts and sciences, Judge Irvine of the Law School, and L. A. Osborne, vice-president of the Westinghouse Electric & Manufacturing Company. The arrangements for the occasion were in charge of D. S. Wegg, chairman of the committee and secretary of the branch. A clever program had been prepared for the occasion and each guest was also presented with a number of fine souvenirs, the gift of various manufacturing firms.

On Monday, March 18, the Secretary attended a smoker given by the members of the Society in Pittsburgh and conveyed to them the belief of the Council that the Engineers Society of Western Pennsylvania was adequately providing for the present needs of the membership for meetings in Pittsburgh. He also expressed the desire of the Council and of the Society to coöperate with them in every way.

The Student branch at Pennsylvania State College was visited by the Secretary on March 19. A meeting of the branch was held in the afternoon and in the evening the Secretary addressed the student body on the Relation of the Engineer to the Public.

ELECTIONS TO HONORARY MEMBERSHIP

At the February Meeting of the Council the President announced the election to Honorary Membership of Anatole Mallet, Paris, France; Dr. Carl Gustav Patrick De Laval, Stockholm, Sweden; and Dr. Rudolf Diesel, Munich, Germany.

M. Mallet has been engaged on various mechanical questions since 1867, among which are steam engines with double expansion and the application of this system to locomotives. The first application was realized in 1875 on the locomotive of the Bayonne and Biarritz Railway; and afterwards on the locomotives of railways of Southwest Russia, of Austria, of Spain and Switzerland. Since 1884, M. Mallet has studied the type of articulated compound locomotives, which has been applied to small locomotives with four axles of 12,000 kg. up to the enormous locomotives with 12 axles of which ten are grinding axles and of a weight of 250,000 kg.

Dr. De Laval in his earlier years did important work in developing processes for the manufacture of steel; but is best known through his invention of the centrifugal cream separator and other apparatus for the use of dairies. The necessity of a high-speed motor for driving the cream separator led to the invention of the De Laval steam turbine with its various characteristic features such as the flexible shaft and gear drive. His inventions have materially helped Sweden to become a great agricultural country and have assisted in disposing of its products to other countries.

Dr. Diesel is widely known as the inventor of the Diesel engine, which he described in a paper read before the National Society of Engineers of Germany in 1897, and which has since been taken up in every country, so that today thousands of Diesel engines are in operation, ranging from 20 to 500 h.p. Dr. Diesel is a member of the Franklin Institute and was decorated with the Order of St. Michel in the third Class.

MEETING OF THE COUNCIL

At the meeting of the Council for March a report by A. M. Hunt of San Francisco was presented of the Convention recently called in San Francisco to formulate plans for the organization of an International Engineering Congress in connection with the Panama Pacific Exposition in 1915, and an invitation extended to this Society to hold its Spring Meeting in that year in San Francisco.

The Council passed a resolution accepting this invitation, the meeting to be the regular Semi-Annual Meeting of the Society, and voted that an invitation be extended to representatives of foreign societies of like character to join in this meeting.

It was also voted that a Committee be appointed by the President to take up the subject of standard cross-sections and symbols for refractories, as embodied in the paper of H. De B. Parsons given at the Annual Meeting, and suggested further by Bradley Stoughton of New York.

The Council further voted, in view of an invitation of the New England Water Works Association asking this Society and the National Association of Master Steam and Hot Water Fitters to join with them in extending the standardization of flanges for pipes up to and including eighty-four inches, that the former Flange Committee be reappointed. As a result it is expected that the recent work of this Committee will be extended and a rational basis for further standardization be finally recommended.

CODE OF ETHICS FOR ENGINEERS

The board of directors of the American Institute of Electrical Engineers have adopted a code of ethics for the professional conduct of engineers. It takes up first a statement to the effect that the engineer in his relations should be guided by the highest principles and that he should satisfy himself to the best of his ability that the enterprises with which he becomes identified are of a legitimate character. It then discusses under suitable divisions the engineer's relations to client or employer; the ownership of engineering records and data, considering employer and client, or customer as well as the engineer; the engineer's relation to the public; and the engineer's relations to the engineering fraternity. This statement of principles is to be published in full in the Proceedings of the American Institute of Electrical Engineers.

RESUSCITATION FROM ELECTRIC SHOCK

The first meeting of the Commission on Resuscitation from Electric Shock was held in the board room of the National Electric Light Association, New York, on February 22. This commission was organized upon the initiative of the National Electric Light Association, and has for a purpose the study of electric shock and the preparation of a set of rules for first aid in case of electrical accident. The commission is composed of members of the American Medical

Association, National Electric Light Association and American Institute of Electrical Engineers.

At the meeting in New York City the medical members of the commission unanimously advocated the Schaefer, or prone, method as the best means in the hands of laymen for maintaining respiration in victims of electric shock, and the commission formally voted to recommend this method. A chart is now being prepared which will give details of first aid in cases of electric accidents and will describe fully the method of applying artificial respiration. This chart will be issued under the auspices of the National Electric Light Association.

GEORGE WALLACE MELVILLE

A career which has added luster to the entire engineering profession was brought to a close on March 17, 1912, in the death of Rear-Admiral George Wallace Melville, U. S. N. Retired, at his home in Philadelphia, Pa., after a brief illness.

Born of Scotch parentage illustrious for its bravery, George Wallace Melville was one of the first to volunteer in his country's service at the outbreak of the Civil War. In 1804, James Melville of Sterling, Scotland, had come to New York, bringing with him his family, one of whom, Alexander Melville, a chemist, became the father of the noted engineer. George Wallace, oldest son of Alexander and Sarah Douther Wallace Melville, was born in New York on January 10, 1841, educated in the public schools, and showing early mechanical ability, sent to the Polytechnic School of Brooklyn and, later, to a religious academy noted for its excellent mathematical training. He had entered the engineering works of James Binn in East Brooklyn, and had acquired the foundation for that practical skill which served him so well in many a later event, when war was declared.

Melville was at once appointed an officer of the Engineering Corps of the United States Navy, for which he was well fitted despite his extreme youth, and throughout the war saw almost continuous service. He will perhaps be best remembered for his daring work in connection with the capture of the Confederate cruiser Florida in the neutral waters of Brazil—a breach of international law, but a justifiable and clever strategy.

The war ended, Melville's adventurous spirit soon sought for new foes to conquer, and when in 1873 the sealing steamer Tigress was chartered to rescue the remnant of Captain Hall's last and fatal voyage to the Arctic, Melville volunteered as her engineer officer. The White North proved an antagonist worthy of Melville's bravery and skill, and it was for his share in the long struggle with the enemy, conquered only within the past two years, that his name first became honored throughout the civilized world. When in 1879 the Jean-

nette, commanded by Lieut. George W. De Long, embarked upon an expedition to the North Pole through Bering Strait, hitherto an untried route, Melville was of the staff, although the department reluctantly assented to his assignment because of the difficulty of filling his place. On the voyage he was called by De Long "a tower of strength," "always self-helpful and reliant in emergency," "of indomitable energy," yet ever cheerful and with a smile and word of encouragement for all. "When he does retire," says De Long, in the time of crisis, "he lies awake, planning some new means of pumping a ship by steam which will be more economical than the main boilers."

Such planning was much needed, for soon after leaving California the vessel was caught in the ice and never after escaped from the floe which held her. In 1881, Melville manned a small force sent by De Long to plant the American flag on what is now known as Henrietta Island, which proved a most hazardous undertaking and which has been most graphically described by Melville himself in his book, "In the Lena Delta." After a three days' journey of constant danger and struggle with the moving ice, hindered by various mishaps and by the frightened dogs, the island was triumphantly claimed for the American government. Of this feat De Long says: "If this persistence in landing upon this island, in spite of the superhuman difficulties he encountered, is not reckoned a brave and meritorious action, it will not be from any failure on my part to make it known."

On June 7, 1881, the Jeannette, worn out with the crushing force of the moving ice, went down, leaving her crew upon the floe, the nearest source of food 500 miles away in the Lena Delta, and the nearest known land, the desert New Siberian Islands, 150 miles distant. For forty-one days they toiled on, Melville ever ready to aid the sick and discouraged and full of energy and expedients. At length coming upon an island hitherto unknown, now called Bennett Island, they rested for nine days and then when the Arctic winter was setting in, embarked southward in three boats, commanded respectively by De Long, Chipp and Melville. Separated by a terrific storm, Melville's crew were finally brought to port, alive but in a pitiable condition. Chipp's vessel had gone down in their sight, a victim of the waves; De Long's fate they could only conjecture. Instead of pushing his way southward, "whither," as Bennett says in The Steam Navy of the United States, "a warmer sun was shining and where lay the great Russian road leading to the borders of

civilization, to the railway, to the blue sea, to home," Melville set out to find De Long, who in his own heart he feared had perished. "Few men," says Bennett, "after passing through all the perils that Melville had survived, would have thrown away that seeming last chance of personal safety, but he, being cast in rare, heroic mould, turned his back upon the means of saving himself and set his rugged face to the northward, forcing his way into the darkness and the awful silence of the Arctic winter like one deliberately invading the dominions of death."

How Melville pressed forward, weak and footsore, suffering from cold and exposure, quelling the mutinies of his native companions from time to time, only to fail in his quest; how, when spring came, he went forth undaunted a second time, to discover at last the bodies of the brave De Long and his party and to bury them, as he has himself written, "in the sight of the spot where they fell, the scene of their suffering and heroic endeavor, where the everlasting snows would be their winding sheet, and the fierce polar blasts would wail their wild dirge through all time,"—is now a matter of common knowledge and need not here be more fully recounted.

In the spring of 1884 Melville made his last trip to the North, this time with the Greeley relief expedition, and was one of the first to reach the dying men at Cape Sabine, a fit ending to this chapter of his career.

The Government had not been unmindful of Melville's broad knowledge of engineering and his intrepid spirit, and in 1887 when the new Navy was in process of building, Admiral Melville was appointed Engineer-in-Chief with the relative rank of Commodore.

At this time the four Roach Cruisers had been completed and contracts had been placed for a few of the succeeding vessels. Except for these few, however, the machinery of the new Navy was all designed by Melville or under his supervision until the time of his retirement sixteen years later. Even this does not represent the total amount of designing done by the Bureau under his direction, since the work was so radically new that tentative designs had to be prepared in many cases, which subsequent changes in the general features of the vessel caused to be given up.

Machinery aggregating more than 1,000,000 h.p. was designed and built under his supervision for vessels of all types from battleships to torpedo boats, involving millions of dollars. He was the

first engineer-in-chief to advocate and actually install water tube boilers. The first to be thus fitted out was the Monterey, equipped with Ward boilers of about 5,000 h.p. He also adopted water tube boilers exclusively for sea launches with the best results, but did not allow himself to be carried away unduly with the idea. He felt that the best interests of the Navy required that the first boilers installed be tested thoroughly before adopting this type as a settled policy.

Ever vigilant to keep our Navy in the forefront of progress he consistently demanded highest possible speed in vessels of all types. In 1898, when other nations were building large battleships of 18 knots speed or over, it was proposed by the Navy Department to construct new vessels copied after the slower 16-knot battleships of previous years; but the Admiral's determined stand and the willingness of contractors to furnish ships capable of attaining the higher speed enabled us to get vessels of 18 knots. As another evidence of this policy, reference should be made to the two triple-screw flyers, Columbia and Minneapolis, which were the first very large vessels to be equipped with triple screws, each vessel developing about 20,000 h.p. and breaking the speed record for naval vessels larger than torpedo-boats at that time. This was one of the boldest pieces of designing which has been done in the Navy Department and is but another evidence of the courage and foresight displayed by Melville upon all occasions where progressive action was demanded.

He applied the method of testing the speed of vessels now known as the "Standardized Screw," and at the time of his retirement had completed an extensive investigation of the use of fuel oil on shipboard.

In the war with Spain, bitterly opposed by Melville with all his customary strength of opinion, his foresight and determination were nevertheless responsible for the equipment of Key West as a naval base and the sending out of the Vulcan, the first repair ship to be used in warfare.

In 1899 came the culmination of the long fight which the engineers in the navy had made to secure actual rank as naval officers instead of relative rank with professional titles. Inasmuch as their work has come to be largely executive, involving the command of a considerable portion of the ship's company, the engineers felt that they should have title commensurate with their duties. This was opposed for years by the line officers, but the splendid work of the

Engineer Corps under Melville and his own personality brought about a better feeling and the Personnel Board was appointed, presided over by Assistant Secretary of the Navy Roosevelt. The result of the deliberations of this board was a reorganization of the entire personnel of the navy, which consolidated the engineers with the line. It cannot be doubted that the work and bearing of Melville and the accomplishments of his department were largely responsible for this sweeping change.

In 1903 Admiral Melville retired from the service in accordance with the sixty-two year limit, though still active and vigorous both mentally and physically, and several years later opened a consulting office in Philadelphia in partnership with J. H. Macalpine, the joint inventor with him of the steam turbine reduction gear for ship and other purposes, which has been described in their book upon the subject, "Broadening the Field of the Marine Steam Turbine," published in 1909. Just before the close of his administration an elaborate series of tests on the use of fuel oil was made under his direction by the U. S. Naval "Liquid Fuel" board, and published as a public document.

Throughout his life Melville's inventive genius was constantly in evidence and there is a long list of inventions to his credit during his work as an ordnance engineer which were never patented. In later years he made numerous patented inventions.

Admiral Melville was justly the recipient of many honors. On his way home from the Jeanette expedition he was granted a private audience by the Czar and Czarina of Russia and was made a Knight of St. Stanislaus of the First Class. Congress in 1900 conferred upon him a gold medal for his heroic service in the Arctic regions and advanced him a grade. He was an honorary member of the Royal Swedish Society of Anthropology and Geography, a member of the National Geographical Society of the United States, the Geographical Society of Philadelphia, the Philosophical Society, the Washington Academy, the Naval Architects and Marine Engineers, and an honorary member of Franklin Institute and of the American Society of Civil Engineers. He was subsequently made also an honorary member of the Institution of Naval Architects of Great Britain, and honorary degrees were conferred upon him by a number of colleges. His service in the Civil War made him a member of the Grand Army of the Republic, of the Naval Order of the United States, and of the Military Order of the Loyal Legion, composed of

commissioned officers of the war. By the latter he was shown much honor and was at the time of his death Commander of the Pennsylvania commandery and Commander-in-Chief of the Order. Some years ago the Loyal Legion placed a fine bust of Admiral Melville, done in bronze, in the Philadelphia Museum.

Admiral Melville became a member of the Society in 1893 and was elected to honorary membership at the Spring Meeting of 1910, held at Atlantic City, where he delivered an address on The Engineer's Duty as a Citizen. From 1895 to 1897 he was a vice-president of the Society and in 1898 became its president, filling the office with much distinction, and giving an interesting account of Engineering in the United States Navy, its Personnel and Material, as his presidential address at the Annual Meeting of 1899. When the Society held its meeting in Washington, D. C., in the spring of 1909, one of its prominent features was an address on the Engineer in the Navy delivered by Admiral Melville, followed by the presentation of his portrait, painted by Sigismond de Ivanowski, to the National Gallery. On the occasion of his seventieth birthday, a number of Admiral Melville's friends presented him with a silver platter of fine workmanship and a memorial resolution, handsomely engrossed, which now hang in the rooms of the Society. The Society was represented at Admiral Melville's funeral on March 20, 1912, by Honorary Vice-Presidents Walter M. McFarland, James M. Dodge, Past-President, Coleman Sellers, Jr., and Theo. N. Ely.

A DISCUSSION OF CERTAIN THERMAL PROPERTIES OF STEAM

By G. A. Goodenough

ABSTRACT OF PAPER

The trustworthy data furnished by the Munich experiments on various properties of steam are used as the basis of a general theory. The characteristic equation proposed by Linde is slightly modified and by means of the Clausius thermodynamic relation a formula for specific heat is derived. The resulting specific heat curves are subjected to various tests and the validity of the formula is apparently established. With the aid of the general equation of thermodynamics explicit expressions for the heat content, energy, and entropy of superheated steam are deduced. The values obtained from these formulae are compared with corresponding values in the Marks and Davis table, and substantial agreement is shown.

A DISCUSSION OF CERTAIN THERMAL PROPERTIES OF STEAM

By G. A. Goodenough,[1] Urbana, Ill.
Non-Member

The recent paper [2] by Dr. H. N. Davis on the thermal properties of steam covers the ground so completely that any further contribution may seem superfluous. In one phase of the investigation, however, he acknowledges some lack of success and leaves unsolved a single problem. He says on page 303, "The reconciliation through Clausius' thermodynamic relation of the accepted volume and specific heat measurements in the superheated region is impossible. It is probably the most important of the outstanding problems in the field." It is the purpose of this paper to present an attack on the problem thus suggested.

2 The Clausius relation referred to is expressed by the equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT\left[\frac{\partial^2 v}{\partial T^2}\right]_v$$

The relation is readily derived from the general equations of thermodynamics, and is applicable to all substances. The derivative in the second member is obtained from the characteristic equation $\phi\ (p, v, t) = 0$ of the substance under consideration. If an expression can be found for this derivative from a sufficiently accurate characteristic equation, the rate of variation of specific heat c_p with the pressure is determined; and if sufficient experimental data are available, it should be possible to derive an explicit expression for c_p in terms of the variables p and T. The method has inherent difficulties, as

[1] Professor of Thermodynamics, Univ. of Ill.

[2] Notes on Certain Thermal Properties of Steam, H. N. Davis. Proc. Am. Acad. of Arts and Sciences, vol. 45, pp. 268–311.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

pointed out by Dr. Davis, and so far it has been used with little success.

3 In the following sections an attempt is made to apply the Clausius relation in the case of superheated steam and to reconcile the volume measurements of Knoblauch, Linde, and Klebe with the specific heat measurements of Knoblauch and Jakob, and of Knoblauch and Mollier. The discussion is divided as follows:

a The characteristic equation for superheated steam proposed by Linde is modified without any material sacrifice in accuracy.

b From the modified equation an expression for the specific heat c_p is derived, and the constants are so chosen that the specific heat curves represent quite accurately the results obtained by Knoblauch and Jakob, and Knoblauch and Mollier.

c From the equation for c_p thus developed, explicit equations for the heat content, entropy and energy of superheated steam are derived.

d The values of volume, heat content and entropy deduced from these equations are compared with the values given in the Marks and Davis tables, and substantial agreement is shown.

THE CHARACTERISTIC EQUATION

4 The experiments of Knoblauch, Linde and Klebe furnish reliable data for a characteristic equation connecting the pressure, temperature and volume of superheated steam. To represent these experiments, Linde [1] proposed in the first instance the equation

$$pv = BT - \left[C\left(\frac{373}{T}\right)^3 - D\right]\frac{1}{v} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

but for the sake of convenience in calculation adopted finally the equation

$$pv = BT - p(1+ap)\left[C\left(\frac{373}{T}\right)^3 - D\right] \quad \ldots\ldots\ldots\ldots [2]$$

Linde gives for the constants of [2] the following values (metric units):

$$B = 47.10;\ a = 0.000002;\ C = 0.031;\ D = 0.0052$$

[1] Mitteilungen über Forschungsarbeiten, vol. 21, 1905.

It appears, however, that these constants can be changed somewhat without affecting materially the agreement of the calculated and experimental results. Linde himself calls attention to an objection to the assumed form of the equation which lies in the fact that at a temperature of about 402 deg. cent., the "correction term" of the second member changes sign, and for higher temperatures the fluid becomes a "more than perfect" (übervollkommenes) gas. He suggests in this connection that the objection may be obviated

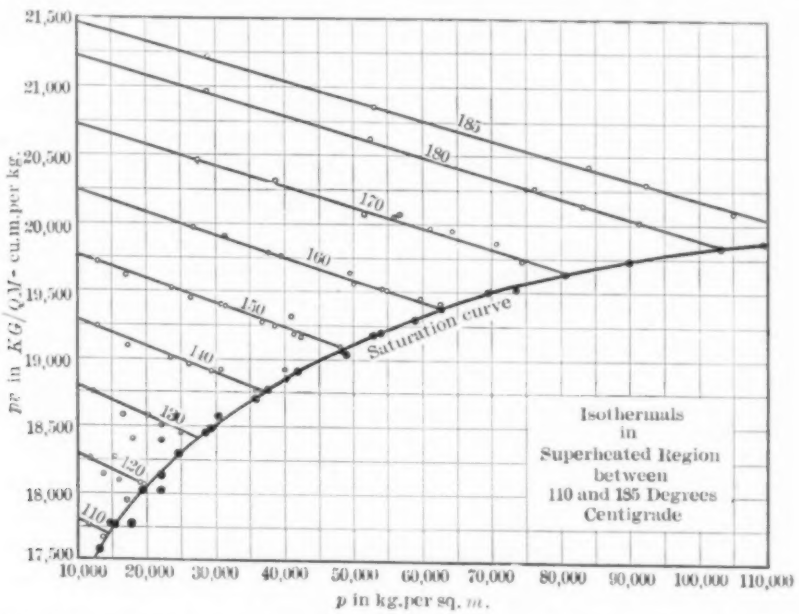

FIG. 1 COMPARISON OF ISOTHERMAL CURVES DEDUCED FROM [3] WITH THE POINTS REPRESENTING THE EXPERIMENTS OF KNOBLAUCH, LINDE AND KLEBE

by increasing the exponent of T from 3 to 3.5, the value assumed by Callendar. The integral value 3 was retained, however, for the sake of convenience in computation.

5 Close inspection of Linde's equation shows that its form is such as to render it somewhat inconvenient for purposes of computation. The essential feature of the formula is the factor $(1+ap)$ which gives the parabolic form to the isothermals when drawn on the pv-p plane. If this be retained, considerable liberty may be taken with the remaining constants without materially affecting the integrity of the equation. In the first place, if the exponent n

be raised from 3 to 3.5, the constant D may be dropped entirely. The equation is then free from Linde's objection that the correction term changes sign at high temperatures, and with constants properly chosen gives results that represent satisfactorily the Munich experiments.

6 In the investigation of other properties of superheated steam, heat content, entropy, etc., considerations appeared that rendered desirable further modification of Linde's equation. The exponent n was raised to 5 and a constant was added to the volume v. The equation therefore took the form:

$$p(v+c) = BT - p(1+ap)\frac{m}{T^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

7 The following are the constants finally chosen for this equation:

Metric Units	English Units
$B = 47.113$	$B = 85.871$, p in lb. per sq. ft.
	$= 0.5963$, p in lb. per sq. in.
$\log m = 11.19839$	$\log m = 13.67938$
$n = 5$	$n = 5$
$c = 0.0055$	$c = 0.088$
$a = 0.00000085$	$a = 0.0006$, p in lb. per sq. in.

8 With these constants equation [3] represents the results of the Munich experiments with substantially the same accuracy as Linde's equation. In fact, when the isothermal curves are drawn on a pv-p chart, as in Fig. 13 of Linde's discussion, the agreement between the curves and experimental points is in some respects better than that given by Linde's equation. The isothermals thus drawn are shown in Fig. 1, which may be compared with Linde's Fig. 13. For a further test of equations [2] and [3], showing that the latter equation may be accepted with confidence at the saturation limit, see Appendix No. 1.

9 As the experiments of Knoblauch, Linde and Klebe extended to only 185 deg. cent., the validity of [3] for higher temperatures is uncertain. It is a safe assertion, however, that [3] is more suitable for extrapolation to high temperatures than is the original Linde equation, but it is possible that equation [3] does not hold at very low temperatures (0 deg. cent.- 40 deg. cent.) in the vicinity of the saturation curve.

FORMULA FOR SPECIFIC HEAT

10 The application of the Clausius relation to equation [3] readily gives the following equation for the specific heat c_p:

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p\left(1+\frac{ap}{2}\right) \quad \ldots\ldots\ldots\ldots [4]$$

or

$$c_p = \phi(T) + f(p, T) \quad \ldots\ldots\ldots\ldots\ldots\ldots [4a]$$

if the term $\frac{Amn(n+1)}{T^{n+1}} p\left(1+\frac{ap}{2}\right)$ is replaced for convenience by $f(p, T)$. For the complete derivation of this equation see Appendix No. 2.

11 If the characteristic equation [3] is correct in form, the expression for the specific heat c_p must have the form given by [4]; and if the arbitrary function $\phi\ (T)$ can be determined, we shall have an explicit expression for c_p in terms of the variables p and T. Callendar and Linde have both used this analysis, starting from their respective characteristic equations, and have thus obtained equations similar to [4].

12 The determination of the proper form of the function $\phi\ (T)$ is the critical point of the whole investigation. If in [4], $p=0$, the equation reduces to $c_p = \phi\ (T)$. Callendar reasons that at zero pressure the specific heat c_p should have an invariable value $(c_p)_o$ independent of the temperature; hence $\phi(T)$ is assumed to be constant. Linde follows Callendar in this assumption. Knoblauch and Jakob, however, on the strength of their direct experiments on the values of c_p, show that $(c_p)_o$ cannot be a constant. It seems possible to establish the form of the function by quite another line of reasoning. Inspection of [4] shows that the correction term $f\ (p, T)$, having the factor T^{n+1} in the denominator, grows rapidly smaller as T increases. At moderately low temperatures ($T=100$ to 600 deg. cent.) this term has an appreciable value, but at temperatures above say 1000 deg. cent., the exponent $n+1$ being 6, the term practically disappears, and the specific heat is given by the relation

$$c_p = \phi\ (T) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

for all pressures. The determination of the function $\phi(T)$ is reduced therefore to the discovery of the relation between specific heat and temperature at high temperatures. The experiments of Mallard and Le Chatelier at temperatures above 3000 deg. cent. and the experiments of Langen at about 1700 deg. cent. agree in making the specific heat a linear function of the temperature, for the temperature range within which the experiments were made. While this linear relation may not, and probably does not, hold for the entire region of superheat, it may be assumed as a close approxi-

mation for a moderate range of temperature. The function $\phi(T)$ therefore takes the simple form

$$\phi(T) = \alpha + \beta T \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [6]$$

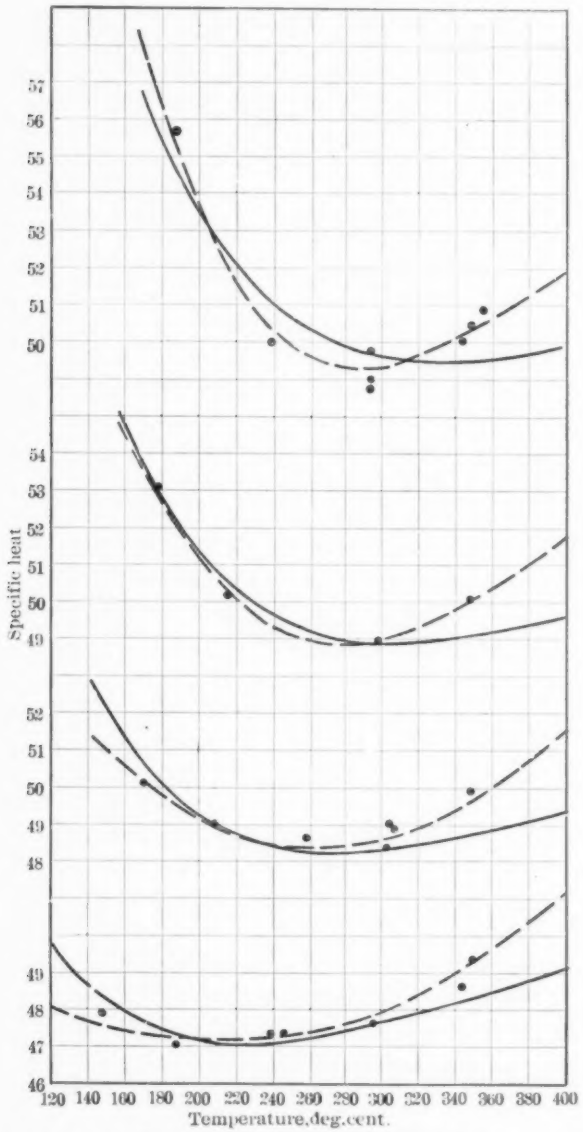

FIG. 2 COMPARISON OF SPECIFIC HEAT CURVES WITH KNOBLAUCH AND JAKOB POINTS. PRESSURES, 2, 4, 6, AND 8 KG. PER SQ. CM.

and equation [4] becomes

$$c_p = \alpha + \beta T + \frac{Amn(n+1)}{T^{n+1}} p\left(1 + \frac{ap}{2}\right) \quad \ldots\ldots\ldots\ldots [7]$$

13 The constants a, m, and n are those heretofore given in connection with the characteristic equation [3], while in the determination of the constants α and β we may utilize the results of the various experiments of Knoblauch and Jakob, and the more recent experiments of Knoblauch and Mollier. For a fuller discussion of the values of these constants see Appendix No. 3.

14 The values finally chosen were:

$$\alpha = 0.367 \qquad \beta = 0.00018$$

Hence

$$\phi\,(T) = 0.367 + 0.00018T \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

or for the fahrenheit scale

$$\left.\begin{aligned} \phi\,(T) &= 0.367 + 0.0001T \\ \phi\,(T) &= 0.413 + 0.0001t \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

The agreement between the curves that represent equation [7], using the expression for $\phi\,(T)$ given by [8], and the Knoblauch and Jakob points is shown in Fig. 2. The agreement is quite satisfactory except for points in the region of 350 deg. The empirical curves adopted by Knoblauch and Jakob (shown in dash lines) were made to pass through these points and were therefore given a sharp slope above 350 deg. Furthermore, at the higher pressures the empirical curves were given much greater slopes at the intersections with the saturation curve than were shown by the curves calculated from [7]. As a result the extrapolated values of c_p at the saturation limit are suspiciously high. Marks and Davis, on the strength of results of Holborn and Henning, lowered the Knoblauch and Jakob curves in the region of high superheat, but accepted their values near saturation.

15 The experiments of Knoblauch and Mollier extend the temperature range to 550 deg. cent. The points representing these experiments are shown in Fig. 3, and the curves exhibited in the same figure represent equation [7] with the values of α and β originally chosen. For the lower pressures the agreement between points and curves is satisfactory. At the higher pressures the agreement could be slightly improved by raising the value of α from 0.367 to 0.369 or 0.370. For further discussion of the validity of equation [7] and values chosen for the constants α and β see Appendix No. 6.

HEAT CONTENT, ENERGY AND ENTROPY

16 With the aid of an explicit formula for the specific heat, the following expressions for the heat content, energy and entropy of superheated steam are readily derived:

For heat content:

$$i = \alpha T + \tfrac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1+\frac{ap}{2}\right) - Acp + i_o \quad \ldots\ldots [10]$$

For intrinsic energy (in b.t.u.):

$$u = T(\alpha + \tfrac{1}{2}\beta T - AB) - \frac{Amp}{T^n}\left[n + (n-1)\frac{ap}{2}\right] + i_o' \quad \ldots\ldots [11]$$

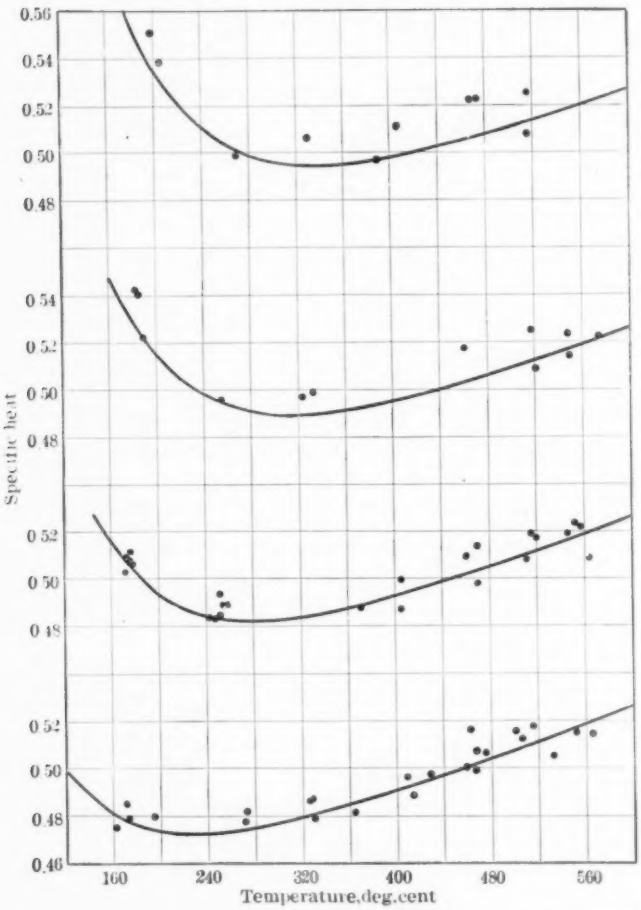

FIG. 3 COMPARISON OF SPECIFIC HEAT CURVES WITH KNOBLAUCH AND MOLLIER POINTS. PRESSURES, 2, 4, 6, AND 8 KG. PER SQ. CM.

For entropy:

$$s = \alpha \log_e T + \beta T - AB \log_e p - Anp\left(1+\frac{ap}{2}\right)\frac{m}{T^{n+1}} + s_o \quad \ldots\ldots [12]$$

For the derivation of these equations see Appendix No. 4.

17 Formula [10] gives the heat content i of superheated steam at any assumed pressure p and temperature T. Since the formula should hold good at the saturation limit, it also gives the value of i for saturated steam when corresponding saturation values of p and T are used. A comparison of the saturation values of i thus obtained with the values given by Marks and Davis will furnish a test of the accuracy of equation [10], and incidentally a test of the entire theory built up from equations [3] and [7]. Taking 1150.4 b.t.u. as the value of i for saturated steam at 212 deg. (the value given by Marks and Davis), the constant i_o is found to be 887.3; and using this value of i_o, the values of i shown in Table 1 are obtained. For a further discussion of the agreement between the two sets of values, and in particular for a proposed modification of Dr. Davis' formula for the heat content see Appendix No. 5.

TABLE 1 VALUES OF i AT SATURATION

Temperature, Deg. Fahr.	Calculated from [10] $i_o = 887.3$	Marks and Davis	Difference, Per Cent
32	1079.6	1073.4	0.58
50	1087.0	1081.4	0.52
75	1097.3	1092.5	0.44
100	1107.4	1103.6	0.34
125	1117.4	1114.5	0.26
150	1127.2	1125.3	0.17
200	1146.0	1145.8	0.02
212	1150.4	1150.4	0.00
250	1163.5	1163.8	−0.03
300	1179.5	1179.1	+0.04
350	1193.6	1191.5	0.18
400	1205.7	1201.3	0.35
450	1215.3	1208.0	0.61

18 From the comparison of the two sets of values in Table 1, and from the considerations set forth in Appendix No. 5, the following conclusions may be drawn:

(*a*) Equation [10] gives with fair accuracy the saturation values of the heat content from 120 deg. to 450 deg. fahr., that is, from a pressure of 1.7 lb. per sq. in. to 420 lb. per sq. in.

(*b*) Equation [10] therefore probably gives quite accurately the heat content of superheated steam for this pressure range and for the degree of superheat usually employed in technical applications.

(*c*) The validity of equation [10] carries with it the validity of the analysis, and justifies the choice of constants in equations [3] and [7].

(*d*) For extremely low pressures the exact values of the various properties of steam are still in doubt, and further investigation is required.

RESUMÉ OF FORMULAE

19 The formulae for the important properties of superheated steam, with the numerical values of the constants inserted, are here collected for convenience of reference. In all cases the constants are so chosen that pressures are to be taken in pounds per square inch.

(*a*) For the specific volume:

$$v = 0.5963\frac{T}{p} - (1+0.0006p)\frac{C}{T^5} - 0.088$$

$$\log C = 13.67938$$

(*b*) For the specific heat:

$$c_p = 0.367 + 0.0001T + p(1+0.0003p)\frac{C_2}{T^6}$$

$$\log C_2 = 14.42408$$

(*c*) For the heat content:

$$\iota = T(0.367+0.00005T) - p(1+0.0003p)\frac{C_3}{T^5} - 0.0163p + 887.3$$

$$\log C_3 = 13.72511$$

(*d*) For the intrinsic energy in b.t.u.:

$$u = T(0.2566+0.00005T) - p(1+0.00024p)\frac{C_4}{T^5} + 887.3$$

$$\log C_4 = 13.64593$$

(*e*) For the entropy:

$$s = 0.8451 \log T + 0.0001T - 0.2542 \log p - p(1+0.0003p)\frac{C_5}{T^6} - 0.3960$$

$$\log C_5 = 13.64593$$

COMPARISON OF RESULTS

20 From the formulae in Par. 19 the properties of superheated steam may be calculated for assumed values of p and T. It may be of interest to compare the values thus obtained with the values given in the Marks and Davis table. Such a comparison is shown in Table 2. Various pressures from 2 to 300 lb. per sq. in. are assumed and the superheat ranges from 0 deg. to 400 deg. fahr.

TABLE 2 COMPARISON OF THE CALCULATED VALUES OF THE PROPERTIES OF SUPERHEATED STEAM WITH THE VALUES GIVEN IN THE MARKS AND DAVIS TABLE

	Pressure, Lb. per Sq. In.																		Superheat, Deg.
	2		15		50		100		150		200		300						
	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal	(M and D)	Cal.	(M and D)	Cal.	(M and D)	Cal.	(M and D)					
Volume, v			26.32																
	174.0	173.5	26.32	26.27	8.52	8.51	4.44	4.43	3.02	3.01	2.30	2.29	1.55	1.55					0
	189.1	189.2	28.41	28.40	9.18	9.19	4.79	4.79	3.26	3.27	2.48	2.49	1.67	1.69					50
	204.2	204.2	30.47	30.46	9.83	9.84	5.12	5.14	3.49	3.51	2.65	2.68	1.79	1.83					100
	234.2	234.2	34.53	34.53	11.07	11.11	5.75	5.80	3.92	3.97	2.98	3.04	2.01	2.09					200
	294.0	293.9	42.55	42.59	13.51	13.58	6.98	7.07	4.74	4.83	3.60	3.71	2.43	2.55					400
Heat Content, i	1117.8	1115.0	1150.7	1150.7	1173.6	1173.6	1187.6	1186.3	1195.9	1193.4	1201.6	1198.1	1209.3	1204.1					0
	1139.8	1137.8	1174.3	1174.2	1199.2	1198.8	1214.8	1213.8	1224.2	1223.0	1231.0	1229.8	1240.2	1240.3					50
	1161.7	1160.7	1197.3	1197.6	1223.9	1223.4	1240.8	1239.7	1251.2	1249.6	1258.7	1257.1	1269.2	1268.2					100
	1206.1	1206.4	1243.8	1244.4	1272.3	1271.8	1291.0	1289.4	1302.7	1300.0	1317.3	1307.7	1323.7	1319.3					200
	1297.5	1298.6	1337.7	1338.3	1368.7	1367.4	1389.6	1385.9	1403.0	1397.4	1412.9	1405.9	1427.6	1418.6					400
Entropy, s	1.9231	1.9180	1.7549	1.7549	1.6578	1.6581	1.6028	1.6020	1.5710	1.5692	1.5490	1.5456	1.5175	1.5129					0
	1.9587	1.9559	1.7887	1.7886	1.6911	1.6909	1.6363	1.6358	1.6049	1.6043	1.5828	1.5823	1.5518	1.5530					50
	1.9921	1.9904	1.8199	1.8199	1.7214	1.7211	1.6665	1.6658	1.6350	1.6343	1.6130	1.6120	1.5822	1.5824					100
	2.0526	2.0529	1.8761	1.8768	1.7759	1.7755	1.7201	1.7188	1.6884	1.6862	1.6662	1.6632	1.6354	1.6323					200
	2.1561	2.1586	1.9730	1.9733	1.8688	1.8671	1.8111	1.8079	1.7781	1.7735	1.7555	1.7493	1.7238	1.7168					400

21 It will be seen that there is substantial agreement between the values for the heat content i and also between the values for the entropy s. The maximum difference between the values of i is slightly over 0.6 of one per cent and that between the values of s is about 0.4 of one per cent. The discrepancy between the values of v is somewhat larger at the higher superheats. From Linde's own statement of the defect in his formula when applied at high temperatures, it appears probable that the values obtained from formula [3] are more accurate than those calculated from Linde's formula. It is remarkable that the values of i and s obtained from quite different specific heat curves should agree so well. And it is gratifying that the accuracy of the Marks and Davis tables is confirmed by this close agreement.

22 Acknowledgment is due Mr. A. L. Schaller and Mr. John A. Dent for assistance in the calculations and for valuable suggestions.

APPENDIX No. 1

VALIDITY OF EQUATIONS [2] AND [3] AT THE SATURATION LIMIT

$$pv = BT - p(1+ap)\left[C\left(\frac{373}{T}\right)^3 - D\right]$$

$$p(v+c) = BT - p(1+ap)\frac{m}{T^n}$$

23 Equations [2] and [3] may also be tested by calculating from them values of v at the saturation limit and comparing these with values determined experimentally or with values calculated from the properties of saturated steam by means of the well known Clapeyron-Clausius formula. Table 3 gives a series of values thus determined.

TABLE 3 SPECIFIC VOLUME v AT SATURATION

Temperature, Deg. Cent.	Experiments of K. L. and K.	Calculated from [2]	Calculated from [3]	Calculated by Henning
100	1.674	1.674	1.673	1.673
120	0.8922	0.8915	0.8915	0.8912
140	0.5091	0.5083	0.5084	0.5078
160	0.3073	0.3070	0.3071	0.3071
180	0.1943	0.1943	0.1945	0.1947

24 The values given in the last column are of special significance. They are those calculated by Henning, using as a basis his recently determined values of the latent heat and Holborn and Henning's accurate values of $\frac{dp}{dt}$. That these values agree so closely with the values determined by the direct experiments of Knoblauch, Linde and Klebe is convincing evidence of the accuracy attained in both sets of experiments. It will be observed that the values of v_{sat} given by [3] agree substantially with both sets of values; hence it appears that equation [3] may be accepted with confidence at the saturation limit.

APPENDIX No. 2

DERIVATION OF EQUATION [4]

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p\left(1 + \frac{ap}{2}\right)$$

25 Writing the characteristic equation [3] in the form

$$v + c = \frac{BT}{p} - (1 + ap)\frac{m}{T^n}$$

the following derivatives are readily obtained:

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}}(1 + ap) \quad \ldots\ldots [13]$$

$$\frac{\partial^2 v}{\partial T^2} = -\frac{mn(n+1)}{T^{n+2}}(1 + ap) \quad \ldots\ldots [14]$$

Substituting the second derivative in the Clausius equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT\frac{\partial^2 v}{\partial T^2}$$

the result is the relation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = \frac{Amn(n+1)}{T^{n+1}}(1 + ap) \quad \ldots\ldots [15]$$

Taking T as a constant and integrating [15] with p as the independent variable, the result is

$$c_p = \frac{Amn(n+1)}{T^{n+1}} p(1 + \frac{ap}{2}) + \text{constant of integration}$$

The constant of integration may be a function of T, since T was held constant during the integration; hence

$$c_p = \phi(T) + \frac{Amn(n+1)}{T^{n+1}} p(1 + \frac{ap}{2}) \quad \ldots\ldots [16]$$

APPENDIX No. 3

CONSTANTS α AND β IN EQUATIONS [6] AND [7]

$$\phi(T)=\alpha+\beta T$$

$$c_p=\alpha+\beta T+\frac{Amn(n+1)}{T^{n+1}}p\left(1+\frac{ap}{2}\right)$$

26 The limits of the value of β are furnished by the linear equations deduced from the explosion experiments and from the experiments of Holborn and Henning. Thus the results of the experiments of Mallard and Le Chatelier at a temperature above 3000 deg. are represented by the equation

$$c_p=0.432+0.000318T$$

Langen's experiments at about 1700 deg. cent. are represented by the equation

$$c_p=0.438+0.000239T$$

and the experiment of Holborn and Henning for temperature below 800 deg. cent. by the equation

$$c_p=0.446+0.0000956T$$

There is reason to believe that the values given by Holborn and Henning are too low. It appears that the coefficient of T is not constant but increases with the temperature; and the value chosen for β will be influenced by the temperature range over which the formula for c_p is assumed to extend. Since in technical applications the temperatures encountered are relatively low, β should probably be smaller than Langen's value 0.000239, and bearing in mind the probable error in the results of Holborn and Henning, β should be considerably higher than the value 0.0000956.

27 In the original determination of the constants α and β, the results of the experiments of Knoblauch and Jakob were used. The later experiments of Knoblauch and Mollier had not been published. The "correction term" $f(p, T)$ of equation [7] was calculated for assumed temperatures and for the pressures 2, 4, 6 and 8 kg. per sq. cm. employed by Knoblauch and Jakob. Then by trial, values of α and β were so chosen as to make the specific heat curves

$$c_p=\alpha+\beta T+f(p, T)$$

fit as closely as possible the experimental points.

APPENDIX No. 4

DERIVATION OF EQUATIONS [10], [11] AND [12]

$$i = \alpha T - \tfrac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1+\frac{ap}{2}\right) - Acp + i_o$$

$$u = T(\alpha + \tfrac{1}{2}\beta T - AB) - \frac{Amp}{T^n}\left[n + (n-1)\frac{ap}{2}\right] + i_o$$

$$s = \alpha \log_e T + \beta T - AB \log_e p - Anp\left(1+\frac{ap}{2}\right)\frac{m}{T^{n+1}} + s_o$$

28 With the aid of an explicit formula for the specific heat, expressions for the heat content, energy, and entropy of superheated steam are as easily derived. For this purpose the general equation

$$dq = c_p dT - AT\left(\frac{\partial v}{\partial T}\right)_p dp \qquad [17]$$

is most convenient. Since the heat content i is defined by the relation

$$i = A(u + pv)$$

we have

$$di = c_p dT - A\left(\frac{\partial v}{\partial T} - v\right) dp \qquad [18]$$

From the characteristic equation [3] the derivative $\frac{\partial v}{\partial T}$ is found to be

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}}(1+ap)$$

Substituting this and also the expression for c_p in [18], the result is

$$di = (\alpha + \beta T)dT + Amn(n+1)p\left(1+\frac{ap}{2}\right)\frac{dT}{T^{n+1}} - \frac{Am(n+1)}{T^n}(1+ap)dp - Acdp$$

The integral of this exact differential is readily found to be

$$i = \alpha T + \frac{1}{2}\beta T^2 - \frac{Am(n+1)}{T^n} p\left(1+\frac{a}{2}p\right) - Acp + i_o \qquad [19]$$

29 The equation for intrinsic energy is derived from the relation

$$Au = i - Apv$$

Substituting for i the expression given by [10], we obtain for the energy expressed in b.t.u.

$$u = T\left(\alpha + \frac{1}{2}\beta T - AB\right) - \frac{Amp}{T^n}\left[n + (n-1)\frac{a}{2}p\right] + i_o \qquad [20]$$

30 For the entropy an expression is readily found from [17]. Thus

$$ds = \frac{dq}{T} = c_p \frac{dT}{T} - A \frac{\partial v}{\partial T} p dp$$

Introducing in this equation the expressions previously derived for c_p and $\frac{\partial v}{\partial T}$, the result is

$$ds = \left(\frac{a}{T} + \beta\right) dT + Amn(n+1)p\left(1 + \frac{a}{2}p\right)\frac{dT}{T^{n+2}} - AB\frac{dp}{p} - \frac{Amn}{T^{n+1}}(1+ap)dp$$

The integral is

$$s = a \log_e T + \beta T - AB \log_e p - Anp\left(1 + \frac{a}{2}p\right)\frac{m}{T^{n+1}} + s_o \quad \ldots\ldots\ldots [21]$$

31 The constants of integration i_o and s_o may be obtained by passing to the saturation limit. Thus let [10] be written

$$i = i' + i_o$$

where i' stands for the variable part of the second member. The constants a, β, m, n, and a being known, i' can be found for any assumed pressure and temperature. Let i' be calculated for a given temperature T_1 and the corresponding *saturation* pressure p_1; from the properties of saturated steam i is known for this same state, and i_o is at once determined by subtraction. In the same way the constant s_o may be determined.

APPENDIX No. 5

THE SATURATION VALUES OF i FROM FORMULA [10] COMPARED WITH THE VALUES GIVEN BY MARKS AND DAVIS, AND A PROPOSED MODIFICATION OF THE DAVIS FORMULA FOR THE HEAT CONTENT

32 For an intermediate range of temperature 200 deg. to 300 deg. (see Table 1) the agreement between the two sets of values is all that could be desired. Even for the range 100 deg. to 400 deg. the discrepancy only slightly exceeds $\frac{1}{2}$ of one per cent. Below 100 deg. and above 400 deg., however, the discrepancy becomes relatively large. It is possible to distribute the difference between the ends and the middle of the temperature range; thus if i_o be taken as 884.2 formula [19] gives values at 32 deg. and 450 deg., about 0.35 per cent larger and at 212 deg. about 0.3 per cent smaller than the corresponding values of Marks and Davis. If these latter values are accepted as final, the agreement of the two sets of values for the range 32 deg. to 450 deg. with a maximum deviation of $\frac{1}{3}$ of one per cent might be taken as evidence of the validity of equation [10]. However, before passing final judgment it will be instructive to examine somewhat in detail the experimental data relative to the heat content of superheated steam.

33 In Fig. 4, which is substantially a reproduction of part of Fig. 5 in the paper of Dr. Davis, the curve represents the values of i calculated from formula [10], with $i_o = 887.3$. For convenience the C scale of temperature is used, and the range is from 60 deg. to 190 deg. cent. The ordinates are the values of $i - i_{100}$ calories and the points are located from the data contained in Tables 1 and 2 of the paper in question. The points indicated by the symbol * represent the measurements of Henning; the other points are those deduced by Dr. Davis from the throttling experiments of Grindley, Griessmann and Peake. In Fig. 5 the curve shows values of i in b.t.u. calculated from equation [10] and the points represent values of i deduced from experiments on the latent heat r by Dieterici, Griffiths, Smith and Henning. The range 32 to 212 deg. fahr. is covered.

34 Directing attention to Fig. 4, it is seen that the points representing Henning's experiments accord well with the points deduced from the throttling experiments with the exception of the points at 140 deg. and 180 deg. Dr. Davis asserts that the point at 180 deg. is not entitled to nearly so much weight as the others because of a defect in the experimental apparatus. Probably this assertion is well founded. On the other hand, the points deduced from the throttling experiments in the region of 180 deg. are perhaps slightly low. In deriving the values from which these points are located, Dr. Davis used the specific heat measurements of Knoblauch and Jakob. Near the saturation limit these values of c_p are undoubtedly too high at the higher pressures. The result of using lower values of c_p is to raise the points in question. The change, however, is slight and is not sufficient to bring the points upon the curve.

35 An examination of the derivative $\frac{di}{dt}$ along the saturation curve may afford another basis of comparison. From the second degree equation of Dr. Davis, namely,

$$i = 1150.4 + 0.3745(t-212) + 0.00055(t-212)^2$$

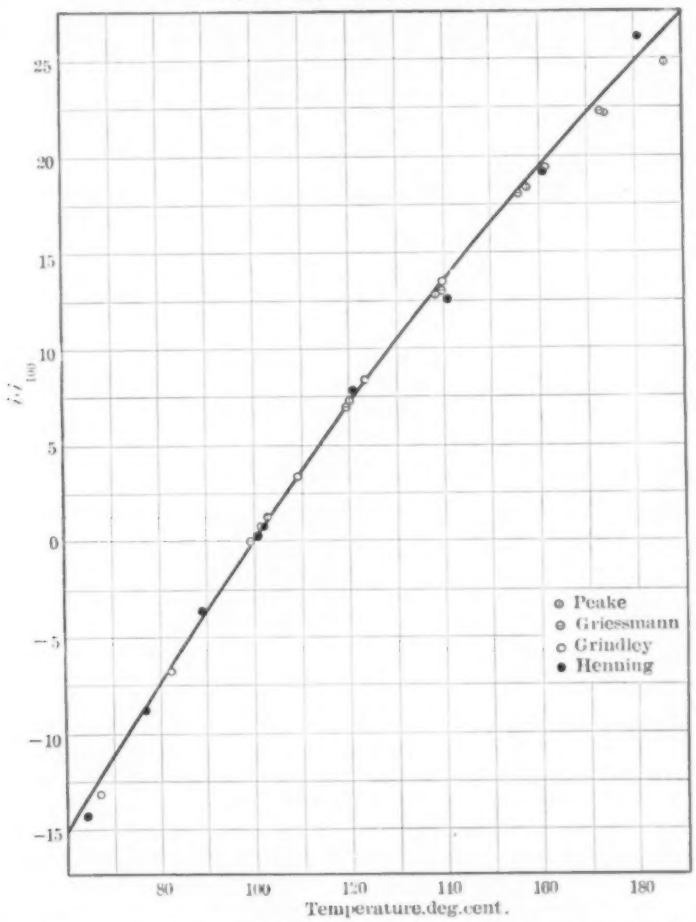

FIG. 4 CURVE REPRESENTING VALUES OF $i-i_{100}$ CALCULATED FROM [10]. THE POINTS GIVE VALUES DETERMINED FROM THE EXPERIMENTS OF HENNING, GRINDLEY, GRIESSMANN AND PEAKE

we have

$$\frac{di}{dt} = 0.3745 - 0.0011(t-212)$$

This linear relation is shown by the straight line A, Fig. 6. Writing [19] in the form $i = (p, T)$, we obtain

$$\left[\frac{di}{dt}\right]_{sat} = \left[\frac{di}{dT}\right]_p + \left[\frac{\partial i}{\partial p}\right]_T \left[\frac{dp}{dt}\right]_{sat} = c_p + \left[\frac{\partial i}{\partial p}\right]_T \frac{dp}{dt} \quad \ldots\ldots\ldots\ldots [22]$$

From [19] an expression for the derivative $\left[\frac{\partial i}{\partial p}\right]_T$ is readily found, and the final result is

$$\left[\frac{di}{dt}\right]_{sat} = c_p - \frac{dp}{dt}\left[\frac{Am(n+1)}{T^n}(1+ap)+c\right] \quad \ldots\ldots\ldots\ldots\ldots [23]$$

The values of this derivative for different temperatures ranging from 200 deg. to 425 deg. fahr. are shown by curve B, Fig. 6. The horizontal straight line C represents the Regnault value $\frac{di}{dt}=0.305$. The form of [23] is such that the relation between $\frac{di}{dt}$ and t cannot be represented by a straight line. Repeated attempts were made to change the constants α, β, m, n and a in such a way as to

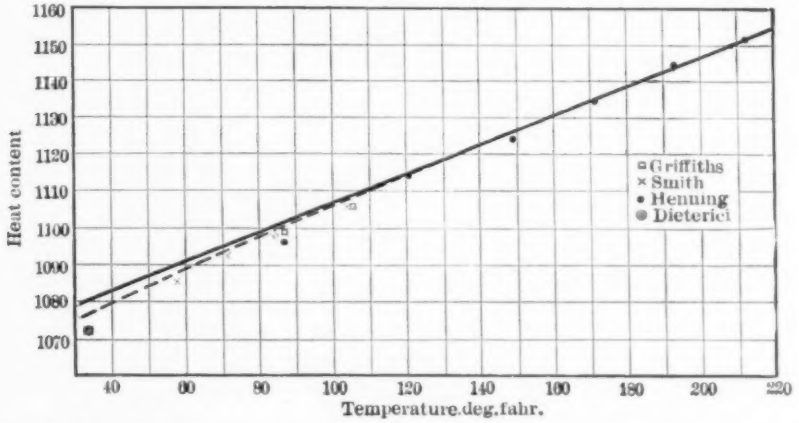

FIG. 5 CURVE REPRESENTING VALUES OF i IN B.T.U. CALCULATED FROM [10]. THE POINTS REPRESENT THE RESULTS OF VARIOUS EXPERIMENTS

straighten curve B and at the same time preserve the integrity of equations [3] and [7]. Such attempts were futile. The result of changing one constant would be neutralized by the necessary change in another, and the position of curve B would remain substantially unchanged. It follows that this curve must be accepted along with the characteristic equation [3] and the specific heat equation [7], or all three must be rejected. Curve B between the temperature limits shown may be approximately represented by a second degree equation though a third degree equation gives a closer agreement. Hence an approximate algebraic equation for i of the form $i=f(t)$ must be of the third degree at least.

36 The formula of Dr. Davis has been so universally accepted that a suggestion that it should be modified may seem presumptuous. The three points brought out in Pars. 33 to 35, however, lead to these conclusions: (*a*) The evidence furnished by Henning's point at 180 deg., whatever value may be attached to it, indicates that the Davis curve should be raised near 180 deg. (*b*) The newer and probably more accurate values of c_p near saturation if used in determining the points deduced from the throttling experiments would raise these points near 180 deg. (*c*) Any equation of the form $i=f(p, T)$ deduced from a characteristic equation and an equation $c_p=\phi(p, T)$, as [10] was deduced,

cannot have a derivative $\left[\frac{di}{dt}\right]_{sat}$ which is a linear function of the temperature. It is perhaps too much to assert that equation [10] is more accurate than the

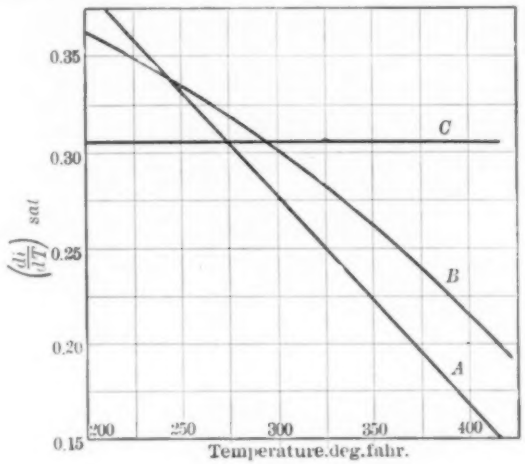

FIG. 6 CURVES FOR $\left(\frac{di}{dT}\right)_{sat} = f(t)$

A from the Davis Equation; *B* from [19]; *C* from Regnault's Equation

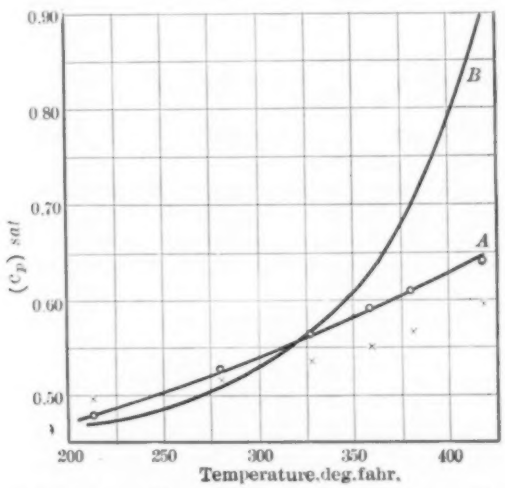

FIG. 7 CURVES SHOWING VARIATION OF $(c_p)_{sat}$ WITH THE TEMPERATURE

Davis formula for temperatures above 212 deg., but it is clear that the discrepancy between the values of i in Table 1, for the higher temperature, may be due in part to errors in the Marks and Davis values of i; it therefore furnishes no adequate basis for a rejection of the analysis leading to equation [10].

37 For temperature below 212 deg. it is seen by reference to Fig. 5 that the curve derived from [19] fits very closely Henning's points from 120 deg. to 212 deg. Below 120 deg. the curve lies somewhat above the points determined by Dieterici, Griffiths, and Smith. It is probable, therefore, that for temperatures below 120 deg. equations [7] and [10] require modification. It may be pointed out that throughout the entire range, equation [10] represents the experiments of Henning about as well as any single empirical formula that could be produced.

38 Equation [21] gives the entropy of superheated steam for assumed values of p and T and may therefore be used to obtain the entropy of saturated steam. It will be of interest to compare the values thus determined with the values obtained in the usual way from the properties of saturated steam. This comparison is shown in Table 4. In the second column are given values from the Marks and Davis table; in the third column values obtained by using in the term

TABLE 4 ENTROPY OF SATURATED STEAM

Temperature, Deg. Fahr.	Entropy at Saturation		
	Marks and Davis	Marks and Davis with Modified Values of r	Calculated from [21]
100	1.9800	1.9869	1.9856
150	1.8674	1.8706	1.8706
200	1.7761	1.7765	1.7766
212	1.7565	1.7565	1.7565
250	1.6996	1.6992	1.6990
300	1.6343	1.6346	1.6342
350	1.5780	1.5800	1.5793
400	1.5286	1.5327	1.5322
450	1.4850	1.4911	1.4911

$\frac{r}{T}$ values of r derived from [10], and in the fourth column values calculated directly from [21]. The constant s_o is found to be -0.3960.

The agreement between the corresponding values in the last two columns scarcely requires comment.

APPENDIX No. 6

VALIDITY OF EQUATION [7] AND VALUES OF CONSTANTS α AND β; ALSO FURTHER TESTS OF THE EQUATION FOR SPECIFIC HEAT

39 The evidence furnished by the Knoblauch and Mollier experiments is clearly in favor of the validity of formula [7] and justifies the values chosen for the constants α and β. The slope of the c_p curves at high temperature is evidently much smaller than was assumed in the original empirical curves of Knoblauch and Jakob and, for the temperature range under consideration, appears to be given correctly by the assumed value of β, namely $\beta = 0.00018$. There is still to be accounted for a considerable discrepancy between the slope thus determined and the slope of the Holborn and Henning curve. Holborn and Henning really determined, not the specific heat of steam directly, but the ratio of this specific heat to the specific heat of air under the same conditions. Comparing the results of Holborn and Henning on the specific heat of air with the results obtained by Joly and Swann, Callendar [1] makes the following statement: "The value found for the mean specific heat of air over the range 115 deg. to 270 deg. cent. by Holborn and Henning was 0.2315, which is about 5 per cent smaller than the probable value over this range. . . . . If the method gives a probable error of 5 per cent over the range 115 deg. to 270 deg., it does not seem at all impossible that the error may amount to 10 per cent over the range 115 deg. to 1400 deg. cent." Callendar's argument seems reasonable, and it is a fair conclusion that the experiments of Knoblauch and Mollier should be accepted as more worthy of confidence than the experiments of Holborn and Henning.

40 Formula [7] may be subjected to several other tests, namely: (*a*) The saturation values of c_p may be checked by Planck's thermodynamic relation; (*b*) the spacing of the c_p curves for different pressures may be tested by a method which connects c_p with the Joule-Thomson coefficient μ. This method is due to Dr. Davis.[2] (*c*) A relation between c_p, μ, and the heat content i of saturated steam affords a check on the values of c_p near the saturation limit at very low temperatures.

41 The values of c_p at saturation may be tested by Planck's equation

$$(c_p)_{sat} = \frac{dH}{dt} - \frac{r}{T} + AT\frac{dp}{dt}\left(\frac{\delta v}{\delta T}\right)_p \quad \ldots\ldots\ldots\ldots\ldots\ldots [24]$$

This equation was used effectively by Dr. Davis in showing that Knoblauch's values are preferable to those given by Thomas. It will be observed that with

[1] Appendix to the Report of the British Association Committee on the Investigation of Gaseous Explosions, Clerk's Gas, Petrol, and Oil Engines, pp. 361–367.

[2] Proc. Amer. Acad., vol. 45, p. 290.

the exception of the derivative $\left[\frac{\partial v}{\partial T}\right]_p$, which may be obtained from [3], the terms in [22] are obtained from the properties of saturated steam. The total heat H and heat content i_{sat} of saturated steam are connected by the relation

$$H = i_{sat} - Apv'$$

in which v' denotes the specific volume of water; hence the derivative $\frac{dH}{dT}$ is found from the relation

$$\frac{dH}{dT} = \left[\frac{di}{dt}\right]_{sat} - Av'\frac{dp}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [25]$$

In the calculation of the values in Table 5 two courses were followed: (*a*) the derivative $\left[\frac{di}{dt}\right]_{sat}$ was found from the Davis equation

$$i = 1150.4 + 0.3745(T-212) - 0.00055(T-212)^2$$

TABLE 5 VALUES OF c_p AT SATURATION

Pressure, Lb. per Sq. In.	Calculated from [22]		Calculated from [7]	Knoblauch and Jakob Curves
	Using Values from Marks and Davis	Using Values from [19]		
15	0.4970	0.4796	0.4775	0.472
50	0.5071	0.5241	0.5228	0.508
100	0.5343	0.5661	0.5604	0.565
150	0.5503	0.5903	0.5876	0.630
200	0.5677	0.6106	0.6097	0.705
300	0.5973	0.6410	0.6454	0.898

and values of $\frac{r}{T}$ were taken directly from the Marks and Davis tables; (*b*) the derivative was found from [10] (see [23], Appendix No. 6) and values of the latent heat r were obtained by subtracting from i_{sat} calculated from [10] the heat content of the liquid given in the Marks and Davis table. Table 5 shows the results of the calculation.

42 In Fig. 7, curve A represents the values of $(c_p)_{sat}$ calculated from [7] and curve B the values given by Knoblauch and Jakob. The points near curve A represent the values in column 3 of Table 5 and the points indicated by crosses represent the values in column 2. The agreement between curve A and the first set of points is sufficiently close to afford a confirmation of equation [7] at the saturation limit. For temperatures above 350 deg. curve B is clearly inadmissible. While the values of $(c_p)_{sat}$ obtained from Planck's equation [24] are in doubt, depending as they do on values of i and r that are not precisely known, it is impossible to bring the points calculated from [24] in harmony with curve B. For temperature above 212 deg., at least, equation [7] satisfies the test furnished by Planck's relation.

43 The specific heat curves derived from equation [7] may be tested by a method developed by Dr. Davis in the paper heretofore cited. It is based upon the following relation between c_p and the Joule-Thompson coefficient

$$c_p = (c_p)_o e^{-\int_p^p \left(\frac{\partial \mu}{\partial t}\right)_p^{dp}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [26]$$

By means of this relation the ratio $\frac{c_p}{(c_p)_o}$ may be calculated for assumed values of p and T along a curve of constant heat content i. The specific heat $(c_p)_o$ is taken at an assumed standard pressure, which was assumed by Dr. Davis as 1 kg. per sq. cm. Evidently the method gives only relative results; that is, it does not give the absolute position of any c_p curve, but rather the spacing of the family of c_p curves. Using the data given by Dr. Davis in Table 5 of his paper, the various ratios $\frac{c_p}{(c_p)_o}$ were calculated from equation [7] and the results obtained were compared with the results derived from the calculations of Dr. Davis. Table 6 shows the two sets of values.

TABLE 6 COMPARISON OF c_p RATIOS

Pressure, Kg. per Sq. Cm.	Curve 1			Curve 2			Curve 3			Curve 4		
	Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_o}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_o}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_o}$		Temperature, Deg. Cent.	$\frac{c_p}{(c_p)_o}$	
		Davis	From [7]		Davis	From [7]		Davis	From [7]		Davis	From [7]
0.1	121.3	0.946	0.945	149.0	0.960	0.962	204.5	0.984	0.981	287.8	0.994	0.993
0.5	121.3	0.970	0.970	150.5	0.979	0.979	205.4	0.991	0.990			
1.0	125.8	1.000	1.000	152.3	1.000	1.000	206.4	1.000	1.000	288.9	1.000	1.000
2.0	130.6	1.060	1.053	155.8	1.041	1.035	208.5	1.018	1.020	290.1	1.007	1.006
3.0	135.1	1.120	1.10	159.1	1.082	1.081	210.5	1.034	1.039			
3.5	137.2	1.150	1.122									
6.0				162.3	1.122	1.107	212.5	1.051	1.057	292.4	1.020	1.023
6.0				168.5	1.200	1.165	216.5	1.084	1.091	294.7	1.032	1.038
8.0				174.3	1.264	1.215	220.2	1.118	1.123	297.0	1.044	1.052
10.0				179.7	1.344	1.260	223.9	1.151	1.151	299.2	1.057	1.066

An inspection of Table 6 shows close agreement at the lower pressures and some divergence at the higher pressures. This test, while of value in a general way, must be taken with some reservation. In the first place the value of the coefficient μ is not known with absolute accuracy. Secondly the constant i curve that must be laid down before the calculation is started depends upon the assumed saturation values of i. If i_{sat} is determined from [10] instead of from the Davis formula, the constant i curves will lie nearer the saturation curve and as a result the values of $\frac{c_p}{(c_p)_o}$ calculated from equation [7] will be larger at the higher pressures.

44 A test that applies at low temperatures near the saturation limit is furnished by Griessman's equation

$$c_p = \frac{\frac{\delta i}{\delta T}}{1 - \mu \frac{dp}{dT}}$$

in which μ denotes the Joule-Thomson coefficient. This relation is in reality identical with that given by equation [22] and may be derived from [22] with the aid of the defining relation $\mu = \left[\frac{\partial T}{\partial p}\right]_i$. Since μ and c_p are intrinsically positive, Griessman's equation imposes the condition that at the saturation limit c_p must be numerically greater than the derivative $\left[\frac{di}{dT}\right]_{sat}$. On page 290 of his paper

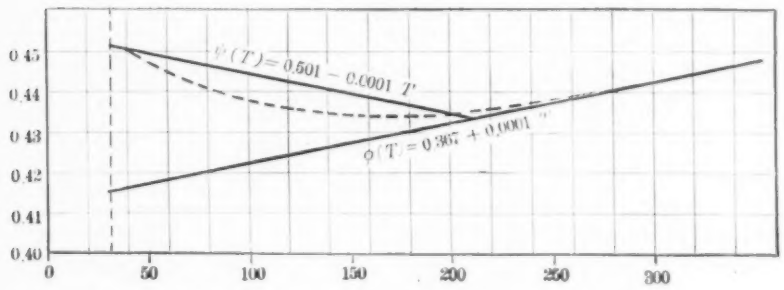

FIG. 8 CURVES REPRESENTING POSSIBLE FORMS OF THE FUNCTION ψ (T) NEAR 32 DEG. FAHR.

Dr. Davis concludes from this condition that at 32 deg. fahr. $(c_p)_{sat}$ must be as great as 0.44. From [7] the values of $(c_p)_{sat}$ is about 0.417. A consideration of this discrepancy throws light on the question heretofore raised relative to heat content at the lower temperatures. The minimum value 0.44 assigned by Mr. Davis is evidently derived from the value of the derivative $\left[\frac{di}{dT}\right]_{sat}$ at 32 deg.; that is, it is the slope of the curve giving the saturation values of i (see Fig. 5) at 32 deg. As the values of i near 32 deg. are very uncertain, no great weight can be attached to this particular number 0.44. From the connection between the equations [7] and [10], Griessman's relation is automatically satisfied at all temperatures by values of c_p calculated from [7] and values of i calculated from [10]; hence at 32 deg. the slope of the curve $i_{sat} = f(T)$, Fig. 5, must be less than 0.417. If, however, the curve were lowered to agree more closely with the experimental points the slope at 32 deg. would be increased, and as a result c_p would be raised. By an arbitrary change in the function $\phi(T)$ within the range of 32 deg. to 212 deg. the curve of Fig. 5 may be wrenched into closer agreement

with the points and $(c_p)_{sat}$ at 32 deg. may be correspondingly raised. In [7] ψ (T) was taken as $0.367+0.0001T$. Let this be replaced by the function $\psi(T) = 0.501-0.0001T$; then in [10] the first two terms for the expression for i become

$$\int \psi(T)dT = 0.501T - 0.00005T^2 + C$$

The straight lines representing $\phi(T)$ and $\psi(T)$ (see Fig. 8) intersect at $t=210$ deg. ($T=670$); hence the constant C is determined by the condition that the value of i for $T=210$ deg. shall be the same when calculated by the modified equation as when obtained from [10]. The effect of this modification of formula [10] is shown by the dash curve, Fig. 5. The resulting value of $(c_p)_{sat}$ at 32 deg. is 0.452, which is very nearly the value taken by Marks and Davis.

45 The process just described is wholly artificial and cannot be justified on any rational basis. There is no reason to suppose that the function $\phi(T)$ is properly represented by a broken line, as shown in Fig. 8. It is conceivable that for the range of temperature under consideration the function should be represented by a curve somewhat of the form shown in the figure; but in the absence of accurate measurements of i or c_p in this region, it is not worth while to complicate the equations by assuming a less simple form for the function $\phi(T)$. The recent work of Smith [1] points to the conclusion that all measurements of the latent heat are probably somewhat in error and that all the points in Fig. 5 are too low; hence the exact value of i near 32 deg. are uncertain. No accurate measurement of $(c_p)_{sat}$ near 32 deg. is available. At present, therefore, we must be satisfied with the following general conclusions:

a The value of $(c_p)_{sat}$ at 32 deg. probably lies between 0.42 and 0.45.

b The curve representing $\phi(T)$ probably lies somewhat above the line $\phi(T)=0.367+0.0001T$ (Fig. 8).

c The curve that represents the relation $i_{sat}=f(T)$ lies very close to the curve of Fig. 5. If Smith's criticism of the latent heat measurement is just, the curve as a whole should be raised.

46 The most important conclusion that may be derived from this discussion is that the analysis is adequate to furnish consistent results when accurate experimental data are available. With accurate measurements of i near 32 deg., the function $\phi(T)$ may be determined and from this accurate value of c_p may be obtained. Conversely, if c_p is accurately determined at 32 deg. and at points in that vicinity, so that the function $\phi(T)$ may be established, then the course of the curve for i_{sat} will be determined.

[1] Physical Review, September 1911.

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE

By Reid T. Stewart

ABSTRACT OF PAPER

A comparison of the theoretical formulae for the strength of steel tubes, pipes, and cylinders under internal fluid pressure shows that:

a Clavarino's formula is theoretically correct for cylinders with attached heads, and Birnie's formula for heads held independently. These formulae are practically applicable to certain classes of seamless steel tubes and cylinders and to critical examination of ordinary commercial steel tubes, pipes and cylinders when sufficiently accurate data are available.

b In commercial welded pipe the variation in thickness of wall, perfection of weld, etc., account for variation in bursting strength of sufficient magnitude to render unnecessary any consideration of Clavarino's or Birnie's condition of head support. All such variations give rise to errors on the side of danger when applying these formulae.

c For ordinary commercial wrought pipe Barlow's formula is to be preferred. This formula is

$$\frac{p}{f}=2\frac{t}{D};\quad p=2f\frac{t}{D};\quad t=\tfrac{1}{2}D\frac{p}{f};\quad f=\tfrac{1}{2}D\frac{p}{t}$$

where D = outside diameter in in.

t = nominal or average thickness of wall in in.

p = internal fluid pressure in lb. per sq. in.

n = safety factor as based on ultimate strength

f = fiber stress in lb. per sq. in.

$=\dfrac{40{,}000}{n}$ for butt-welded steel pipe

$=\dfrac{50{,}000}{n}$ for lap-welded steel pipe

$=\dfrac{60{,}000}{n}$ for seamless steel tubes

$=\dfrac{28{,}000}{n}$ for wrought-iron pipe

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE

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PART I. A COMPARISON OF THE THEORETICAL FORMULAE

In order to arrive at some definite conclusion as to what formula or formulae should be used for calculating the strength of tubes, pipes and cylinders subjected to internal fluid pressure, the different published formulae have been investigated and compared. These are five in number, namely, the common formula, and those by Barlow, Lamé, Clavarino, and Birnie.

2 These formulae have been put into the simplest form for application to tubes, pipes and cylinders, and are reduced to a common notation for the sake of making an easy comparison. The notation used is as follows:

D_1 = outside diameter in inches.
D_2 = inside diameter in inches.
t = thickness of wall in inches.
p = internal gage pressure, or difference between internal and external fluid pressures, in lb. per sq. in.
f = fiber stress in the wall in lb. per sq. in.

3 The formulae here given are for the usual conditions of practise, namely, where the external pressure is atmospheric and the internal pressure is expressed as gage pressure. They are also applicable to cases where the external pressure is not excessive by taking p as the difference between the internal and external pressures.

4 In all that follows it is assumed that the length of the tube or pipe relative to its diameter is sufficiently great to eliminate the influence of end support tending to prevent rupture.

5 *Nature of Stress in Tube Wall.* An internal fluid pressure may give rise (*a*) to a circumferential stress within the wall of a tube or pipe, or (*b*) to both a circumferential and a longitudinal stress acting

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jointly. In either case the tube wall is under radial compressive stress as indicated by the arrows, Figs. 1 and 2.

6 Fig. 1 illustrates a tube with frictionless plungers fitted into its ends, the plungers being kept in place by the external forces, *PP*, which exactly balance the internal fluid pressure tending to force them outward. In this case the tube wall is subjected only to the internal forces shown as acting at right angles to its inner surface. It is obvious that these forces can give rise to radial and circumferential stresses only in the tube wall. The value of the circumferential stress, f_t, in lb. per sq. in. is

$$f_t = p\frac{D_2}{D_1 - D_2} = \frac{pD_2}{2t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

7 Fig. 2 illustrates the ordinary case of a tube or pipe with both ends closed. In this case the tube wall, as in Fig. 1, is subjected to the circumferential stress, f_t, along with the radial stress, and at the same time is subjected to the longitudinal stress, f_l. The longitudinal

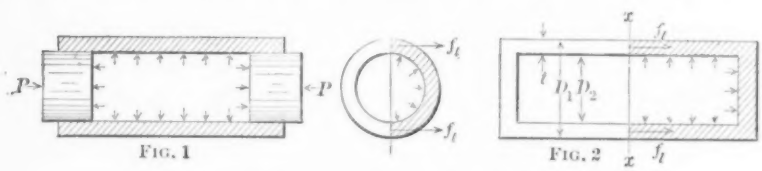

FIG. 1

FIG. 2

FIG. 1 TUBE WITH FRICTIONLESS PLUNGERS
FIG. 2 TUBE WITH BOTH ENDS CLOSED

stress is caused by the internal fluid pressure tending to force the attached heads outward and expressed in lb. per sq. in. is

$$f_l = p\frac{D_2^2}{4(D_2 + t)t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

When the thickness of wall t, is relatively small with respect to the diameter, the longitudinal stress becomes approximately

$$f_l = \frac{pD_2}{4t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

or one-half the corresponding circumferential stress.

8 *Common Formula.* This is the formula generally found in books on mechanics. It is based on the condition that the tube wall is subjected to circumferential stress only, Fig. 1, and assumes (*a*) that the material of the tube wall is devoid of elasticity, and (*b*) that the stress is the same on all the circumferential fibers from the innermost to the outermost. These assumptions are only approxi-

mately true for tubes of comparatively thin walls, and are greatly in error for tubes having very thick walls.

9 Using the notation as given above, the formula is

$$\frac{p}{f}=2\,\frac{t}{D_2};\quad p=2\,f\,\frac{t}{D_2};\quad t=\tfrac{1}{2}\,D_2\,\frac{p}{f};\quad f=\tfrac{1}{2}\,D_2\,\frac{p}{t}\qquad\text{[4]}$$

10 Referring to the charts, Figs. 3 and 4, it will be seen that the common formula gives quite close results for comparatively thin walls when used for the conditions shown in Fig. 1, for which Birnie's formula is theoretically correct. The error increases as the thickness of wall becomes relatively greater, reaching 10 per cent for a thickness ratio $\frac{t}{D_1}$, of about 0.05. For thick walls the error is great; for example, when $\frac{t}{D_1}=0.25$ the value of $\frac{p}{f}$ is about 100 per cent in error. It should be observed when applying the common formula to this case that the error is always on the side of danger.

11 For the conditions shown in Fig. 2, that is, when the tube is subjected to the stresses due to an internal fluid pressure acting jointly on the tube wall and its closed ends, for which Clavarino's formula is theoretically correct, the curves show for a thickness ratio, $\frac{t}{D_1}$, less than 0.07 that the common formula errs on the side of safety, the greatest error being about 12 per cent; while for thickness ratios greater than 0.07 the error is on the side of danger, reaching 10 per cent for a thickness ratio of 0.1 and about 100 per cent for a ratio of 0.25.

12 *Barlow's Formula.* This formula assumes (*a*) that because of the elasticity of the material, the different circumferential fibers will have their diameters increased in such a manner as to keep the area of cross-section constant; and (*b*) that the length of the tube is unaltered by the internal fluid pressure. As neither of these assumptions is theoretically correct, this formula can give only approximately correct results. Using the notation given above, this formula is

$$\frac{p}{f}=2\,\frac{t}{D_1};\quad p=2\,f\,\frac{t}{D_1};\quad t=\tfrac{1}{2}\,D_1\,\frac{p}{f};\quad f=\tfrac{1}{2}\,D_1\,\frac{p}{t}\qquad\text{[5]}$$

13 It should be observed that while Barlow's formula is similar in form to the common formula, it gives results that are quite different when applied to tubes, pipes, and cylinders having walls of considerable thickness. This is due to the fact that Barlow's formula

is expressed in terms of the outside diameter, D_1, whereas the common formula is expressed in terms of the inside diameter, D_2.

14 Referring to Figs. 3 and 4, it will be seen that Barlow's for-

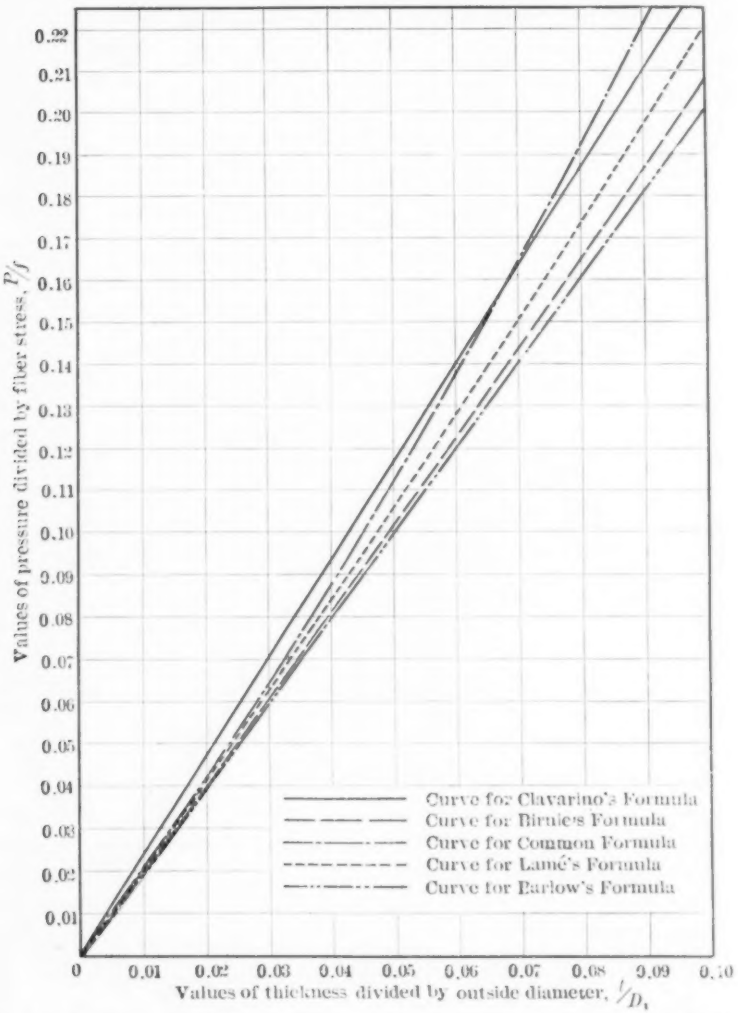

FIG. 3 COMPARISON OF INTERNAL FLUID PRESSURE FORMULAE FOR TUBES, PIPES AND CYLINDERS

mula gives quite close results when used for the condition shown in Fig. 1, for which Birnie's formula is theoretically correct. The curves show for the entire practical range of thickness ratios that

the error in values of $\frac{p}{f}$, for this case, does not exceed 3 per cent, the error throughout the whole practical range being on the side of

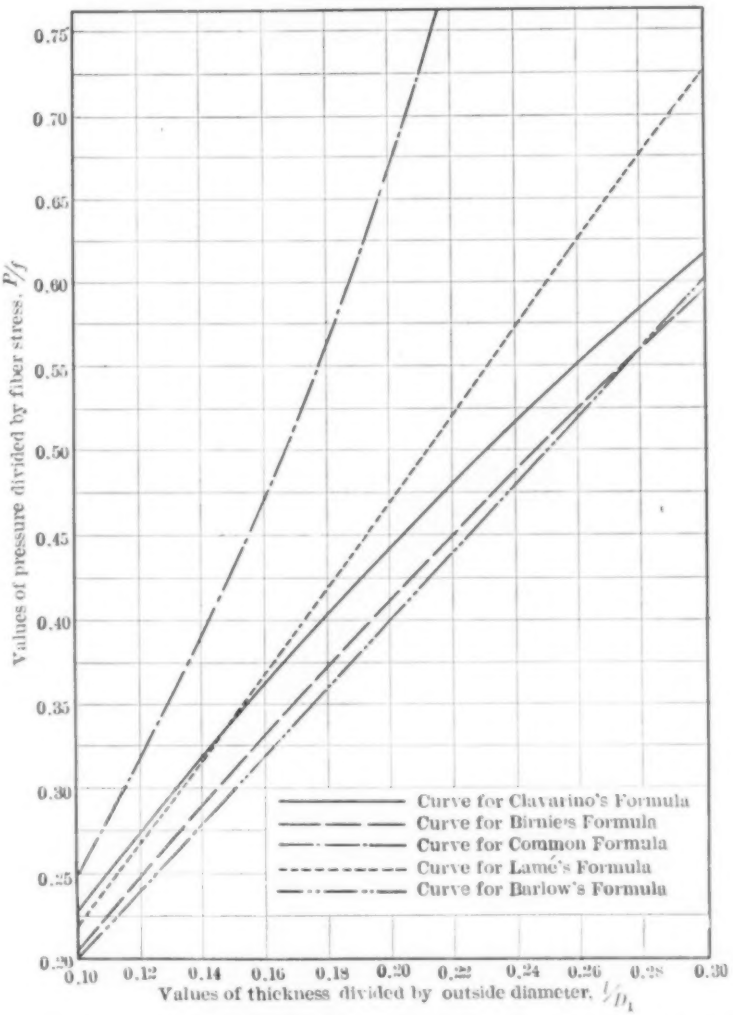

FIG. 4 COMPARISON OF INTERNAL FLUID PRESSURE FORMULAE FOR TUBES, PIPES AND CYLINDERS

safety. This then is the best of the simple theoretical formulae for application to the cases illustrated in Fig. 1.

15 For the conditions shown in Fig. 2, namely when the tube is subjected to the stresses due to an internal fluid pressure acting jointly on the tube wall and its closed ends, for which Clavarino's formula is theoretically correct, the curves show that Barlow's formula gives values of $\frac{p}{f}$ whose errors range from 15 per cent for tubes, pipes and cylinders having thin walls, to 10 per cent for those having thick walls, the error being on the side of safety for all practical thickness ratios.

16 *Lamé's Formula.* This formula is meant to apply to the conditions shown in Fig. 2. Each material particle of the tube wall is supposed to be subjected to the radial compression, the circumferential and longitudinal tensions due to an internal fluid pressure acting jointly on the tube wall and its closed ends; and the material of the tube wall is supposed to be elastic under these actions. Lamé's formula, however, ignores the coefficient of lateral contraction, known as Poisson's Ratio, and consequently is not theoretically correct.

17 Using the notation as given above this formula is

$$\frac{p}{f}=\frac{D_1^2-D_2^2}{D_1^2+D_2^2};\quad p=\frac{D_1^2-D_2^2}{D_1^2+D_2^2}f;\quad D_2=D_1\sqrt{\frac{f-p}{f+p}};\quad D_1=D_2\sqrt{\frac{f+p}{f-p}}\quad\ldots[6]$$

18 Referring to Figs. 3 and 4, it will be seen that Lamé's formula, which is meant to apply to the conditions for which Clavarino's formula is theoretically correct, gives for thickness ratios, $\frac{t}{D_1}$, less than 0.15, an error on the side of safety, the error having a maximum value of about 14 per cent when $\frac{t}{D_1}$ equals 0.01. For thickness ratios greater than 0.15 the error is on the side of danger, reaching 10 per cent for a ratio of about 0.23.

19 *Clavarino's Formula.* In this formula, as in Lamé's formula, each material particle of the tube wall is supposed to be subjected to the radial compression and the circumferential and longitudinal tensions due to an internal fluid pressure acting jointly on the tube wall and its closed ends; and the material is supposed to be elastic under these actions. Unlike Lamé's formula, however, this formula expresses the true stresses in the tube wall as based upon the coefficient of lateral contraction, known as Poisson's ratio, and is consequently theoretically correct for the conditions shown in Fig. 2,

providing the stress on the most strained fiber does not exceed the elastic limit of the material.

20 Using the notation given above and assuming the value of the coefficient of lateral contraction for tube steel to be 0.3, this formula is

$$\frac{p}{f}=\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2};\quad p=\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2}f;\quad D_1=D_2\sqrt{\frac{10f+4p}{10f-13p}};$$

$$D_2=D_1\sqrt{\frac{10f-13p}{10f+4p}}\qquad\qquad[7]$$

21 This theoretically correct formula for the conditions shown in Fig. 2 has the disadvantage that it is difficult to apply directly in making calculations. In order to remove this difficulty Table 1 has been prepared, by means of which any desired calculation can be as readily made by Clavarino's formula as by any of the simpler formulae. The entries of this table are the values in Clavarino's formula of the factor

$$\frac{10(D_1^2-D_2^2)}{13D_1^2+4D_2^2}=K$$

22 It will be observed that these factors are tabulated for thickness ratios, $\frac{t}{D_1}$, from 0.01 to 0.3, advancing by thousandths. Thus for a wall thickness, t, of 0.25 in. and an outside diameter, D_1, of 10 in., the thickness ratio, $\frac{t}{D_1}$, would be 0.25 divided by 10, or 0.025. The required factor corresponding to this thickness ratio is 0.0587 and is found in the column headed 0.005 opposite 0.02 in column 1. Similarly for an outside diameter of 4 in. and a wall thickness of 0.5 in., the thickness ratio would be 0.125 and the corresponding internal pressure factor is 0.2869.

23 If we designate the value of any tabular factor by K, then it is obvious that Clavarino's formula may be written

$$\frac{p}{f}=K;\quad p=Kf;\quad f=\frac{p}{K}\qquad\qquad[8]$$

24 Table 1 is well adapted to the ready solution of problems involving the strength and safety of a tube, pipe or cylinder which is subjected to the stresses due to an internal fluid pressure acting jointly on its wall and closed ends, as illustrated in Fig. 2.

Problem 1. Required the safe working fluid pressure, p (Fig. 2), when the outside diameter, $D_1=4$ in.; thickness of wall, $t=0.5$ in.; and the working fiber

stress of the steel, $f = 10{,}000$ lb. Solution: (*a*) the thickness ratio, $\frac{t}{D_1} = 0.125$; (*b*) the corresponding tabular factor, K, is found from Table 1 to be 0.2869; and (*c*) the required safe working fluid pressure, $p = Kf$, equation [8], or $0.2869 \times 10{,}000$, or 2869 lb. per sq. in.

TABLE 1 INTERNAL FLUID PRESSURE FACTORS, K, FOR CONDITIONS SHOWN IN FIG. 2

CALCULATED BY CLAVARINO'S FORMULA, ASSUMING FOR STEEL A COEFFICIENT OF LATERAL CONTRACTION (POISSON'S RATIO) = 0.3.

RULE: DIVIDE THICKNESS OF TUBE OR PIPE BY ITS OUTSIDE DIAMETER, BOTH BEING EXPRESSED IN INCHES, THEN MULTIPLY THE TABULAR VALUE CORRESPONDING TO THIS QUOTIENT BY THE WORKING FIBER STRESS IN LB. PER SQ. IN. THE RESULT WILL BE THE SAFE INTERNAL PRESSURE IN LB. PER SQ. IN.

$\frac{t}{D_1}$	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
0.01	0.0235	0.0259	0.0282	0.0306	0.0329	0.0352	0.0376	0.0399	0.0423	0.0446
0.02	0.0470	0.0493	0.0517	0.0540	0.0564	0.0587	0.0610	0.0634	0.0657	0.0681
0.03	0.0704	0.0727	0.0751	0.0774	0.0797	0.0821	0.0844	0.0867	0.0891	0.0914
0.04	0.0937	0.0961	0.0984	0.1007	0.1031	0.1054	0.1077	0.1100	0.1123	0.1147
0.05	0.1170	0.1193	0.1216	0.1239	0.1263	0.1286	0.1309	0.1332	0.1355	0.1378
0.06	0.1401	0.1424	0.1448	0.1471	0.1494	0.1517	0.1540	0.1563	0.1586	0.1609
0.07	0.1632	0.1655	0.1678	0.1700	0.1723	0.1746	0.1769	0.1792	0.1815	0.1838
0.08	0.1861	0.1883	0.1906	0.1929	0.1952	0.1974	0.1997	0.2020	0.2043	0.2065
0.09	0.2088	0.2111	0.2133	0.2156	0.2178	0.2201	0.2223	0.2246	0.2269	0.2291
0.10	0.2314	0.2336	0.2358	0.2381	0.2403	0.2425	0.2448	0.2470	0.2493	0.2515
0.11	0.2537	0.2559	0.2582	0.2604	0.2626	0.2648	0.2670	0.2692	0.2715	0.2737
0.12	0.2759	0.2781	0.2803	0.2825	0.2847	0.2869	0.2890	0.2912	0.2934	0.2956
0.13	0.2978	0.3000	0.3022	0.3043	0.3065	0.3087	0.3108	0.3130	0.3152	0.3173
0.14	0.3195	0.3216	0.3238	0.3259	0.3281	0.3302	0.3323	0.3345	0.3366	0.3388
0.15	0.3409	0.3430	0.3451	0.3472	0.3494	0.3515	0.3536	0.3557	0.3578	0.3599
0.16	0.3620	0.3641	0.3662	0.3683	0.3704	0.3724	0.3745	0.3766	0.3787	0.3808
0.17	0.3828	0.3849	0.3869	0.3890	0.3910	0.3931	0.3951	0.3972	0.3992	0.4013
0.18	0.4033	0.4053	0.4073	0.4094	0.4114	0.4134	0.4154	0.4174	0.4194	0.4214
0.19	0.4234	0.4254	0.4274	0.4294	0.4314	0.4333	0.4353	0.4373	0.4393	0.4412
0.20	0.4432	0.4452	0.4471	0.4490	0.4510	0.4529	0.4548	0.4568	0.4587	0.4606
0.21	0.4626	0.4645	0.4664	0.4683	0.4702	0.4721	0.4740	0.4758	0.4777	0.4796
0.22	0.4815	0.4834	0.4852	0.4871	0.4889	0.4908	0.4926	0.4945	0.4964	0.4982
0.23	0.5001	0.5019	0.5037	0.5055	0.5073	0.5091	0.5109	0.5127	0.5145	0.5163
0.24	0.5181	0.5199	0.5216	0.5234	0.5252	0.5269	0.5287	0.5304	0.5322	0.5340
0.25	0.5357	0.5374	0.5391	0.5408	0.5426	0.5443	0.5460	0.5477	0.5494	0.5511
0.26	0.5528	0.5545	0.5561	0.5578	0.5594	0.5611	0.5628	0.5644	0.5661	0.5677
0.27	0.5694	0.5710	0.5726	0.5742	0.5758	0.5774	0.5790	0.5806	0.5822	0.5838
0.28	0.5854	0.5870	0.5885	0.5901	0.5916	0.5932	0.5947	0.5963	0.5978	0.5994
0.29	0.6009	0.6024	0.6039	0.6054	0.6069	0.6084	0.6099	0.6114	0.6129	0.6143
0.30	0.6158	0.6173	0.6187	0.6201	0.6216	0.6230	0.6244	0.6259	0.6273	0.6287

Problem 2. Required the fiber stress, f, in the wall of a cylinder, Fig. 2, when the outside diameter, $D_1 = 5.5$ in., the thickness of wall, $t = 0.25$ in.; and the working fluid pressure, $p = 1500$ lb. per sq. in. Solution: (*a*) the thickness ratio, $\frac{t}{D_1} = 0.045$; (*b*) the corresponding tabular factor, K, is found from Table 1 to be 0.1054; and (*c*) the required fiber stress, $f = \frac{p}{K}$, equation [8], or $1500 \div 0.1054$ or 14,200 lb. per sq. in.

Problem 3. Required the thickness of wall, t (Fig. 2), when the outside diameter, $D_1 = 8$ in.; the working fiber stress of the steel, $f = 15{,}000$ lb. per sq. in.; and the working fluid pressure, $p = 2000$ lb. per sq. in. Solution: (*a*) the factor, $K = \frac{p}{f}$, equation [8], or $2000 \div 15{,}000$, or 0.133; (*b*) the value of the thickness ratio, $\frac{t}{D_1}$, corresponding to this value of K is found from Table 1 to be 0.057;

TABLE 2 INTERNAL FLUID PRESSURE FACTORS, K, FOR CONDITIONS SHOWN IN FIG. 1

CALCULATED BY BIRNIE'S FORMULA, ASSUMING FOR STEEL A COEFFICIENT OF LATERAL CONTRACTION (POISSON'S RATIO) = 0.3.

RULE: DIVIDE THICKNESS OF TUBE OR PIPE BY ITS OUTSIDE DIAMETER, BOTH BEING EXPRESSED IN INCHES, THEN MULTIPLY THE TABULAR VALUE CORRESPONDING TO THIS QUOTIENT BY THE WORKING FIBER STRESS IN LB. PER SQ. IN. THE RESULT WILL BE THE SAFE INTERNAL PRESSURE IN LB. PER SQ. IN.

$\frac{t}{D_1}$	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
0.01	0.0201	0.0221	0.0241	0.0261	0.0282	0.0302	0.0322	0.0342	0.0363	0.0383
0.02	0.0403	0.0423	0.0444	0.0464	0.0485	0.0505	0.0525	0.0546	0.0566	0.0586
0.03	0.0607	0.0627	0.0648	0.0668	0.0689	0.0709	0.0730	0.0750	0.0771	0.0791
0.04	0.0812	0.0832	0.0853	0.0873	0.0894	0.0915	0.0935	0.0956	0.0976	0.0997
0.05	0.1018	0.1038	0.1059	0.1080	0.1100	0.1121	0.1142	0.1163	0.1183	0.1204
0.06	0.1225	0.1245	0.1266	0.1287	0.1308	0.1329	0.1349	0.1370	0.1391	0.1412
0.07	0.1433	0.1453	0.1474	0.1495	0.1516	0.1537	0.1558	0.1579	0.1599	0.1620
0.08	0.1641	0.1662	0.1683	0.1704	0.1725	0.1746	0.1767	0.1787	0.1808	0.1829
0.09	0.1850	0.1871	0.1892	0.1913	0.1934	0.1955	0.1976	0.1997	0.2018	0.2039
0.10	0.2059	0.2080	0.2101	0.2122	0.2143	0.2164	0.2185	0.2206	0.2227	0.2248
0.11	0.2269	0.2290	0.2311	0.2332	0.2353	0.2374	0.2395	0.2416	0.2437	0.2457
0.12	0.2478	0.2499	0.2520	0.2541	0.2562	0.2583	0.2604	0.2625	0.2646	0.2667
0.13	0.2688	0.2708	0.2729	0.2750	0.2771	0.2792	0.2813	0.2834	0.2854	0.2875
0.14	0.2896	0.2917	0.2938	0.2959	0.2979	0.3000	0.3021	0.3042	0.3062	0.3083
0.15	0.3104	0.3125	0.3145	0.3166	0.3187	0.3208	0.3228	0.3249	0.3270	0.3290
0.16	0.3311	0.3332	0.3352	0.3373	0.3393	0.3414	0.3434	0.3455	0.3476	0.3496
0.17	0.3517	0.3537	0.3558	0.3578	0.3598	0.3619	0.3639	0.3660	0.3680	0.3700
0.18	0.3721	0.3741	0.3761	0.3782	0.3802	0.3822	0.3842	0.3863	0.3883	0.3903
0.19	0.3923	0.3943	0.3963	0.3983	0.4003	0.4024	0.4044	0.4064	0.4084	0.4104
0.20	0.4124	0.4144	0.4163	0.4183	0.4203	0.4223	0.4243	0.4262	0.4282	0.4302
0.21	0.4322	0.4341	0.4361	0.4380	0.4400	0.4419	0.4439	0.4459	0.4478	0.4498
0.22	0.4517	0.4536	0.4556	0.4575	0.4594	0.4613	0.4633	0.4652	0.4671	0.4690
0.23	0.4710	0.4729	0.4748	0.4767	0.4785	0.4804	0.4823	0.4842	0.4861	0.4880
0.24	0.4899	0.4918	0.4936	0.4955	0.4973	0.4992	0.5010	0.5029	0.5048	0.5066
0.25	0.5085	0.5103	0.5121	0.5139	0.5157	0.5176	0.5194	0.5212	0.5230	0.5248
0.26	0.5266	0.5284	0.5302	0.5320	0.5338	0.5355	0.5373	0.5391	0.5409	0.5427
0.27	0.5444	0.5462	0.5479	0.5496	0.5514	0.5531	0.5548	0.5566	0.5583	0.5600
0.28	0.5617	0.5634	0.5651	0.5668	0.5685	0.5702	0.5718	0.5735	0.5752	0.5769
0.29	0.5786	0.5802	0.5818	0.5835	0.5851	0.5867	0.5884	0.5900	0.5916	0.5933
0.30	0.5949	0.5965	0.5981	0.5996	0.6012	0.6028	0.6044	0.6059	0.6075	0.6091

and (*c*) the required thickness will result from multiplying this thickness ratio, $\frac{t}{D_1}$, by the outside diameter, D_1, or $0.057 \times 8 = 0.456$ in.

Note—When the inside diameter, D_2, the internal pressure, p, and the working fiber stress, f, are given and it is required to find the thickness of wall, t, proceed

by finding first the value of the outside diameter, D_1, by means of equation [7]; after which the required thickness may be had by taking one-half the difference of the outside and inside diameters or

$$t = \frac{D_1 - D_2}{2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [9]$$

25 *Birnie's Formula.* This formula is based upon the conditions illustrated in Fig. 1. In its derivation precisely the same assumptions are made as for Clavarino's formula, with the single exception that the longitudinal stress, f_1, due to the internal fluid pressure acting upon attached heads, is assumed not to exist. Birnie's formula consequently is theoretically correct for tubes, pipes and cylinders that are subjected to an internal fluid pressure in such manner as not to give rise to longitudinal stress in the wall; provided the stress on the most strained fiber does not exceed the elastic limit of the material.

26 Using the same notation as before and assuming the value of the coefficient of lateral contraction for steel to be 0.3, this formula is

$$\frac{p}{f} = \frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2}; \quad p = \frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2} f; \quad D_1 = D_2 \sqrt{\frac{10f + 7p}{10f - 13p}};$$

$$D_2 = D_1 \sqrt{\frac{10f - 13p}{10f + 7p}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

27 Birnie's formula, like Clavarino's formula, has the disadvantage of being difficult to apply directly in making calculations. In order to remove this difficulty Table 2 has been prepared, the entries being the values in Birnie's formula of the factor

$$\frac{10(D_1^2 - D_2^2)}{13D_1^2 + 7D_2^2} = K$$

28 This table is used in a manner precisely similar to the table of factors for Clavarino's formula.

PART II. RESULTS OF TESTS ON COMMERCIAL TUBES AND PIPES, AND THE APPLICABILITY OF THE THEORETICAL FORMULAE

29 In Part I there appears a full statement of the basis of each of the five theoretical formulae for the strength of tubes, pipes and cylinders, when subjected to internal fluid pressures, together with a comparison of results obtained by their use. One or the other of these formulae, taken apparently at random, has often been used without sufficient understanding of their application to practical conditions. It is the purpose of what follows to illustrate the proper application of these formulae, making use of the results of hydro-

static tests recently made on commercial pipes at one of the mills of the National Tube Company.

30 *Yield Point Tests on Commercial Pipe.* Tests were made under Clavarino's condition, Fig. 2, on 195 specimens of 10-in. and 279 specimens of 12-in. lap-welded steel pipes, all of which were made up into cylinders with heads welded to the pipe. The hydrostatic pressure was raised until the yield point of the material was reached. The unit stresses on the most strained fibers were then calculated by means of Clavarino's formula, the pipes having been measured by micrometer before welding in the head, to determine the least thickness of wall.

31 The average results of the yield points of the most strained fibers of the material constituting these pipes, when compared with the average yield point of tensile test specimens cut from about 400 similar pipes, may be summarized as follows:

Outside diameter of pipe, in	10.00	12.00
Least thickness of wall, in	0.172	0.164
Hydrostatic pressure at yield point, lb. per sq. in	1435	1195
Yield point by Clavarino's formula, lb. per sq. in	35,600	37,100
Yield point average of tensile tests, lb. per sq. in	37,000	37,000
Apparent error in yield point by Clavarino's formula, per cent	−3.8	+0.3

32 This summary of the average results of 474 tests is a very satisfactory confirmation of the accuracy of Clavarino's formula when applied to commercial steel pipes for the conditions under which the formula theoretically applies.

33 Other tests show that when the heads are attached to the pipe, as in Fig. 2, it lengthens upon application of an internal fluid pressure and that when the heads are held independently, as in Fig. 1, it shortens in accord respectively with the assumptions which constitute the basis of Clavarino's and Birnie's formulae regarding change of length under internal fluid pressure.

34 *Applicability of Clavarino's and Birnie's Formulae.* The above summary of results of tests on pipes shows that Clavarino's formula is applicable to commercial wrought-steel pipe for the conditions shown in Fig. 2, when the yield point of the most strained fiber is not exceeded and the least thickness of wall is accurately known.

35 Tests made at the Watertown Arsenal in 1892, 1893, 1894, 1897 and 1902 on sections of steel guns, show that Birnie's formula for the condition shown in Fig. 1, when applied up to the elastic limit of the most strained fiber, gives results which agree with the results of direct tests that are within the ordinary range of experi-

mental error. These Watertown Arsenal tests were all made on tubes the material and dimensions of which were uniform to a degree obtainable only by boring and turning from forgings of the choicest portion of selected ingots.

TABLE 3-a BURSTING TESTS OF COMMERCIAL TUBES AND PIPES

Size	No. of Pieces Burst	Nominal External Diameter, In.	Average Thickness of Walls, In.	Bursting Pressure Pounds per Square Inch			Head Condition*	No. Burst by Failure of Material not at Weld	Average Fiber Stress by Barlow's Formula	Class of Material
				Minimum	Maximum	Average				
Steel, Butt-Welded										
⅛	10	0.405	0.066	11840	17320	14266	*C*	1	44011	Standard pipe
¼	10	0.540	0.085	8830	14680	12206	*C*	1	38645	Standard pipe
⅜	10	0.675	0.088	5850	13030	10330	*C*	1	39272	Standard pipe
½	10	0.840	0.101	11380	16310	14038	*C*	0	58163	Standard pipe
¾	10	1.050	0.109	7150	9150	8020	*C*	0	38657	Standard pipe
1	10	1.315	0.131	4500	8800	6990	*C*	0	35085	Standard pipe
1¼	10	1.660	1.139	4400	7300	5808	*C*	0	34603	Standard pipe
1¼	15	1.660	0.140	5500	11900	7700	*C*	1	45215	Redrawn
1½	10	1.900	0.143	3000	6100	4960	*C*	0	33031	Standard pipe
2	11	2.375	0.149	3830	6060	4951	*C*	0	40485	Standard pipe
2½	10	2.875	0.198	4310	5740	5134	*C*	0	37351	Standard pipe
3	10	3.500	0.204	4650	6370	5398	*C*	0	46234	Standard pipe
1¼	10	1.660	0.180	7910	14280	10514	*C*	0	48922	Extra strong
2	10	2.375	0.213	7250	8940	8238	*C*	0	45935	Extra strong
2	10	2.375	0.220	6160	8920	7661	*C*	0	41347	Extra strong
2	10	2.375	0.445	8500	18314	14992	*C*	0	40023	X X strong
							General average		41686	
Steel Lap-Welded										
2	10	2.375	0.155	4890	7940	6645	*C*	1	50962	Standard pipe
2	10	2.375	0.182	4860	10060	7361	*C*	0	47889	Standard pipe
3	10	3.500	0.210	3830	8200	6368	*C*	7	53560	Standard pipe
4	10	4.500	0.232	4810	5680	5249	*C*	1	51462	Standard pipe
5	10	5.563	0.258	3410	5260	4538	*C*	1	48882	Standard pipe
6	5	6.625	0.275	2450	5210	4088	*C*	0	49286	Standard pipe
6	5	6.625	0.275	3170	4760	3666	*B*	0	44106	Standard pipe
10	5	10.750	0.349	3560	4730	4290	*C*	1	66080	Standard pipe
10	5	10.750	0.347	2770	3940	3396	*B*	2	52692	Standard pipe
2	10	2.375	0.218	2500	9870	7909	*C*	0	43254	Extra strong
2	10	2.000	0.108	5100	6560	6062	*C*	7	55607	Boiler tubes
3	10	3.000	0.112	3220	4860	3967	*C*	1	52957	Boiler tubes
4	5	4.000	0.135	3640	4070	3840	*C*	2	56978	Boiler tubes
4	5	4.000	0.136	3720	4040	3914	*B*	1	57440	Boiler tubes
							General average		52225	

* *C* = Clavarino conditions, Fig. 2. *B* = Birnie conditions, Fig. 1.

36 It is apparent that any variation below the nominal or average value in strength of material, thickness of wall and efficiency of joint in welded pipe, or above the nominal in diameter, will give results which err on the side of danger when making use of either Clavarino's

or Birnie's formulae. These formulae then should be restricted in their use to certain classes of seamless tubes and cylinders and to critical examinations of ordinary tubes, pipes and cylinders, when exact results are desired and sufficiently accurate data are available.

37 For all ordinary calculations of strength of commercial tubes, pipes and cylinders, Barlow's simple approximate formula, equation [5], is preferable.

38 *Bursting Tests of Commercial Tubes and Pipes.* Tables 3-a, 3-b and 4 show the average results of several hundred tests of commercial tubes and pipes, all of which were burst by hydrostatic pressure at one of the mills of the National Tube Company. Of these 95 per cent was made by this company, while 86 per cent of the wrought-iron pipe tested was obtained by purchase in the open market.

TABLE 3-b BURSTING TESTS OF COMMERCIAL TUBES AND PIPES

Size	No. of Pieces Burst	Nominal External Diameter, In.	Average Thickness of Walls, In.	Bursting Pressure Pounds per Square Inch			Head Condition*	No. Burst by Failure of Material not at Weld	Average Fiber Stress by Barlow's Formula	Class of Material
				Minimum	Maximum	Average				
Steel, Seamless										
2	10	2.000	0.098	5420	6590	6052	*C*	10	61530	Boiler tubes
3	10	3.000	0.112	3940	4730	4272	*C*	10	57075	Boiler tubes
4	6	4.000	0.134	4160	4440	4318	*C*	6	64450	Boiler tubes
4	4	4.000	0.134	4250	4440	4328	*B*	4	64488	Boiler tubes
							General average		61886	
Iron, Butt-Welded										
1¼	10	1.660	0.136	2880	6290	5283	*C*	3	32126	Standard pipe
1¼	10	1.660	0.136	3640	5680	4891	*C*	1	29817	Standard pipe
2	10	2.375	0.156	2930	4250	3687	*C*	2	28051	Standard pipe
1¼	10	1.660	0.188	2770	7330	5895	*C*	1	26678	Extra strong
							General average		29168	
Iron, Lap-Welded										
2	10	2.375	0.152	2400	3940	3213	*C*	1	25122	Standard pipe
2	10	2.375	0.207	5530	7120	6349	*C*	8	36461	Extra strong
							General average		30792	

* *C* = Clavarino conditions, Fig. 2. *B* = Birnie conditions, Fig. 1

39 The average ultimate tensile strength of pipe steel is 57,000 lb. per sq. in., whether taken in the direction of rolling or transversely thereto, while that of the seamless steel tested is 60,000 lb. per sq. in. No tensile tests were made of the material of the wrought-iron pipes.

40 An examination of these tables will lead to the following general conclusions:

a In commercial welded pipe the variations in thickness of

TABLE 4 STRENGTH OF WELDS OF COMMERCIAL TUBES AND PIPES SELECTED FROM PRECEDING TABLE OF BURSTING TESTS

Size	No. Pieces Bursting in Weld[1]	Average Fiber Stress by Barlow's Formula	Class of Material
Steel, Butt-Welded			
⅜	9	43938	Standard pipe
¼	9	37777	Standard pipe
⅝	9	38954	Standard pipe
½	10	58163	Standard pipe
¾	10	38657	Standard pipe
1	10	35085	Standard pipe
1¼	10	34603	Standard pipe
1¼	14	45643	Redrawn
1½	10	33031	Standard pipe
2	11	40485	Standard pipe
2½	10	37351	Standard pipe
3	10	46234	Standard pipe
1¼	10	48922	Extra strong
2	10	45935	Extra strong
2	10	41347	Extra Strong
2	10	40023	X X Strong
General average		41634	
Steel, Lap-Welded			
2	9	50052	Standard pipe
2	10	47889	Standard pipe
3	3	54510	Standard pipe
4	9	51019	Standard pipe
5	9	48852	Standard pipe
6	10	47026	Standard pipe
10	7	59537	Standard pipe
2	10	43254	Extra Strong
2	3	56933	Boiler tubes
3	9	51980	Boiler tubes
4	7	57521	Boiler tubes
General average		51688	
Iron, Butt-Welded			
1¼	7	31136	Standard pipe
1¼	9	30680	Standard pipe
2	8	27323	Standard pipe
1¼	9	27073	Extra strong
General average		29053	
Iron, Lap-Welded			
2	9	24581	Standard pipe
2	2	34340	Extra strong
General average		29461	

[1] These only are included in averages.

wall, perfection of weld, etc., give rise to variations in bursting strength of sufficient magnitude to render unnecessary any consideration of Birnie's or Clavarino's

condition of head support as shown in Figs. 1 and 2 respectively.

b The relative strengths of steel pipes and tubes, when using Barlow's formula and basing the calculations on average diameter, thickness of wall and ultimate tensile strength of material, are as follows: For butt-welded steel pipe, 73 per cent; for lap-welded steel pipe, 92 per cent; and for seamless steel tubes, approximately 100 per cent. In steel pipe, then, the strength of the butt-weld is about 80 per cent of that of the lap-weld.

c The relative strengths of wrought-iron and steel pipe, from Table 3, are as follows: Butt-welded wrought-iron pipe is 70 per cent as strong as similar butt-welded steel pipe; and lap-welded wrought-iron pipe is 57 per cent as strong as similar lap-welded steel pipe.

41 *Applicability of Barlow's Formula.* Of the five formulae considered in Part I, that by Barlow is the best suited for all ordinary calculations pertaining to the bursting strength of commercial tubes, pipes and cylinders. The theoretical error on the side of safety resulting from its use will generally not exceed the actual combined error on the side of danger when using either Birnie's or Clavarino's formula, due to the ordinary range of variation in the thickness of wall, strength of the material, etc., when applied to the ordinary commercial product. This is true, at least up to the yield point of the material, for any ratio of thickness of wall to outside diameter less than three-tenths. In this respect Barlow's formula is very superior to the common approximate formula which gives errors that are absurdly large on the side of danger for very thick walls (Fig. 4).

42 For all ordinary calculations, then, pertaining to the bursting strength of commercial tubes, pipes and cylinders use Barlow's formula [1] (equation [5]).

$$\frac{p}{f}=2\frac{t}{D_1};\quad p=2\,f\,\frac{t}{D_1};\quad t=\tfrac{1}{2}\,D_1\,\frac{p}{f};\quad f=\tfrac{1}{2}\,D_1\,\frac{p}{t}$$

where

D_1 = outside diameter, in.

[1] For certain classes of seamless tubes and cylinders and for critical examinations of welded pipe, where the least thickness of wall, yield point of material, etc., are known with accuracy, and close results are desired, see Clavarino's formulae and Birnie's equations [7] and [10].

t = average thickness of wall, in.
p = internal fluid pressure, lb. per sq. in.
n = safety factor as based on ultimate strength
f = working or safe fiber stress, lb. per sq. in.

$= \frac{40{,}000}{n}$ for butt-welded steel pipe

$= \frac{50{,}000}{n}$ for lap-welded steel pipe

$= \frac{60{,}000}{n}$ for seamless steel tubes

$= \frac{28{,}000}{n}$ for wrought-iron pipe

43 These average values of f are based on the accompanying tables of bursting tests of commercial tubes and pipes. They are intended for substitution in Barlow's formula in case more exact data for the working fiber stress are not at hand.

EQUIPMENT OF A MODERN FLOUR MILL ON A GRADUAL REDUCTION SYSTEM

By John F. Harrison and W. W. Nichols

ABSTRACT OF PAPER

The paper contains a description of a modern flour mill of 4000 bbl. capacity in 24 hours. Detail drawings show the arrangement of machinery, also the drive and history of the development in the art of milling, describing the revolution in the process of manufacturing flour that has taken place in America within the last 40 years. Accompanying figures show the exterior and interior of a mill and elevator of the latest type. The system of a gradual reduction roller mill is also outlined.

EQUIPMENT OF A MODERN FLOUR MILL ON A GRADUAL REDUCTION SYSTEM

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The art of milling is one of the most ancient, having been employed by all tribes or nations from the first stages of existence. The mills of the ancients differed little from those at present in vogue in the East and in rural Mexico. They consist of two circular stones, the lower being fixed, having its upper surface made convex so as to fit into a concavity of the upper stone containing a central hole through which the grain is fed to the grinding surface. The upper stone has a pin or handle near the outer edge with which to turn it. This type of mill has remained unchanged in form for from 4000 to 5000 years.

2 An improvement on the ancient hand-mill was made by furrowing, with a system of grooves, the surface of the stones to give a grinding action, at the same time discharging the meal at the periphery of the stones, and the application of animal power to operate them. Public water-mills were mentioned in Roman laws and used in Britain by the Romans. The sluice of one was discovered in Lancashire, in the middle of the last century. This stage in the art of milling brings us to the old grist mill with one run of stone and a sifting reel, generally operated by water power or windmills when the former was not available, and then to the large mills of Europe and America containing several run of stone and a number of reels with machinery for the cleaning of wheat. Milling up to this time had in reality made little progress from ancient times.

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

3 Oliver Evans, a Colonial Delaware millwright, improved and invented machinery and appurtenances of mills which revolutionized the manufacture of flour. His system of conveyors and elevators made the mill in its operation practically automatic and gave the first impetus to improvement in the mills in this country. In Europe Oliver Evans' mills were referred to as "scientific milling" machines.

4 At the conclusion of the Napoleonic wars when men turned their thoughts to the arts and sciences of peace, the art of milling received its share of attention. Larger mills were erected, water power developed, improvements made in machinery for the cleaning of wheat and the process made more automatic, but still on the lines of the old millstone and the hexagon sifting reel. Merchant mills centered where hydraulic power was available, as at Budapest in Hungary and Richmond, Oswego, Rochester, Milwaukee, St. Louis and Minneapolis in America, these cities each in their turn becoming milling centers but little in advance, as regards engineering, of the Oliver Evans mills.

5 In 1874 Minneapolis mills had a daily capacity of 4500 bbl. of flour, with 190 run of stone, as opposed to the present capacity of 85,000 bbl., one mill alone having 14,000 bbl. daily output.

6 The Washburn mill "A" which rivaled in size and equipment those of Budapest, in Hungary, had 40 run of stone with a daily capacity of 1200 bbl. of flour. About the time of its erection began the most important revolution in the art of milling which has developed the present system and mechanical construction, and the enterprise of the millers of Minnesota in adapting these inventions influenced and encouraged the rapid settlement of the northwestern states of Minnesota and the Dakotas. While the two most important machines, the roller mill and the middlings purifier, as well as the new process of milling, commonly called the Hungarian system, were of European origin, the adapting and improvement in the line of milling machinery and perfecting of the system is very largely the work of American milling engineers.

7 This evolution was carried on in the face of persistent opposition by a conservative class of men who were experts in the old style, a very essential feature of which was the care of the millstone, skill in the dressing and balancing of which required mechanical ability and practice. At the present time this is almost a lost art. By the old system the object was to reduce the wheat to flour at one grinding by a mashing or squeezing process and to obtain the largest amount of flour with the least amount of middlings, since these

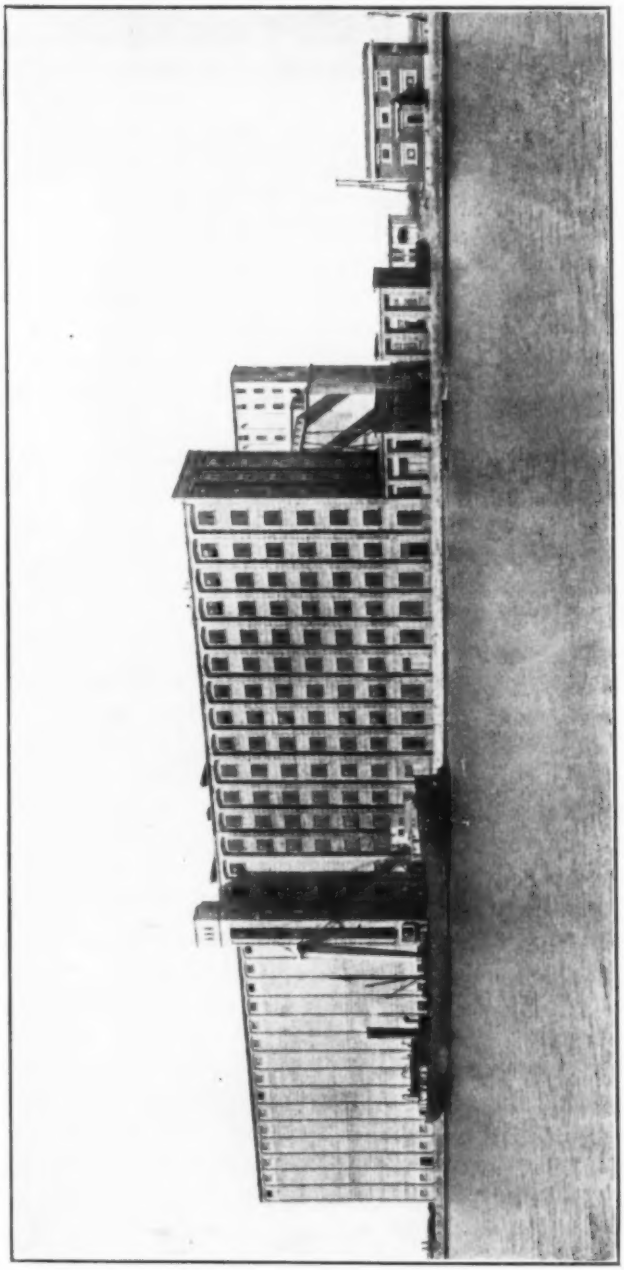

FIG. 1 FRONT VIEW OF MILL BUILDING AND MOTOR HOUSE

contained fine particles of bran and shorts and were available only for a low grade of flour. The new or Hungarian process reversed the old system and ground so as to make as little flour as possible, reducing the wheat to granules which were sifted in a reel clothed with graded silk, thus separating the flour from grits or middlings and bran. The introduction of the middlings purifier made it possible to separate the fine particles of fluff and bran and regrind the purified middlings, producing a higher grade of flour.

8 This new process advanced the price of the hard spring wheat of Minnesota which formerly sold in Eastern markets for 50 cents per bushel less than winter wheat, and Minnesota flour brought from $1 to $2 less per bbl. than the brands manufactured by the St. Louis mills which used the softer varieties of winter wheat. The mills of Minnesota using the new process according to the Chicago market reports of February 1874, sold "strictly fancy" New Process flour at $9.50 to $10.50 per bbl. while choice winter wheat St. Louis flour brought in the same market $8.25 and $9.25 per bbl.

9 The middlings purifiers were used in Europe on the hard varieties of Hungarian wheat some years before their introduction in America. They were of a rather primitive pattern at first, but improved step by step until the Hungarian miller had developed a very elaborate machine. The purification was effected by grading the middlings on sieves, as many as ten separations being made, the offal and fluff being taken off by a suction of air in aspirators. In Hungary the purification of middlings was carried further than was thought necessary in America where a much simpler system requiring less labor to operate was developed.

10 M. Lacroix, a French miller, introduced a middlings purifier into a Minneapolis mill in the early sixties. The construction was a sieve about 30 in. wide, 12 to 16 ft. long, shaken laterally by an eccentric shaft. The box containing the sieve was fitted with a suction fan drawing the air through the silk, the air being admitted through openings in the frame under the silk, which was graded to sift the fine middlings at the head and the coarse toward the tail, the coarse particles of shorts tailing over. The silk required brushing by hand quite frequently to prevent clogging. The application of a traveling brush under the silk was suggested and patented. This was the most successful purifier of the many afterwards put on the market and is the standard middlings purifier today (Fig. 8), displacing the Hungarian machine, not only in America but also to a large degree in Europe.

11 In connection with the purifier a settling chamber or dust room was used which occupied a large space of the mill. The dust settled on the floor of the room, and the air inseparable from the fine dust discharged into the open air. This was not only a crude system but wasteful and dangerous, the fine flour dust under certain conditions being as explosive as gunpowder. On May 2, 1878, the large Washburn "A" mill referred to in Par. 6 was destroyed by an explosion of flour dust, leveling the huge stone structure with foundation walls 6 ft. thick and literally leaving not one stone upon another.

12 This necessitated the present dust collector, an important American invention shown in Fig. 8, which collected the dust in a number of cloth tubes. The air passes through the cloth and the tubes are cleaned by jarring them over a dead air chamber.

13 The roller mill, one of the most important machines in a modern flour mill, was invented about 1833 in Switzerland. For 30 years after their first introduction, however, they were used in but few mills in Europe. The first rolls introduced in Minneapolis in 1874, were crude machines made of cast iron and much secrecy was used in connection with the experiments made. One process of flattening the wheat before grinding was soon abandoned. The fluting of the rolls to grind or scrape the flour from the bran proved a success as it enabled the miller to grind higher and produce more middlings; this was the first step towards gradual reduction. The cast iron being too soft to last, a fluted chilled-iron roll was successfully manufactured and in a short time displaced the millstone for this purpose. The application of fluted rolls for breaking or grinding the wheat was gradually adopted, five to eight operations being made in the gradual reduction of the wheat. After each grinding or break, the product was sent to a reel or scalper, the flour separated and the middlings sent to a purifier while the tailings were delivered to the next reduction on the rolls, the final separation of bran being cleaned in a duster.

14 Oscar Oexle, an engineer and millwright of Augsburg, Bavaria, in 1877 introduced the Wegner porcelain roller mill in America and with it a system of reducing the middlings on porcelain rolls in connection with millstones. Smooth chilled-iron rolls being more durable displaced these porcelain rolls.

15 For some years roller mills were used indiscriminately on different stock in different mills in connection with millstones and to a great extent were an experiment. A complete roller mill, the

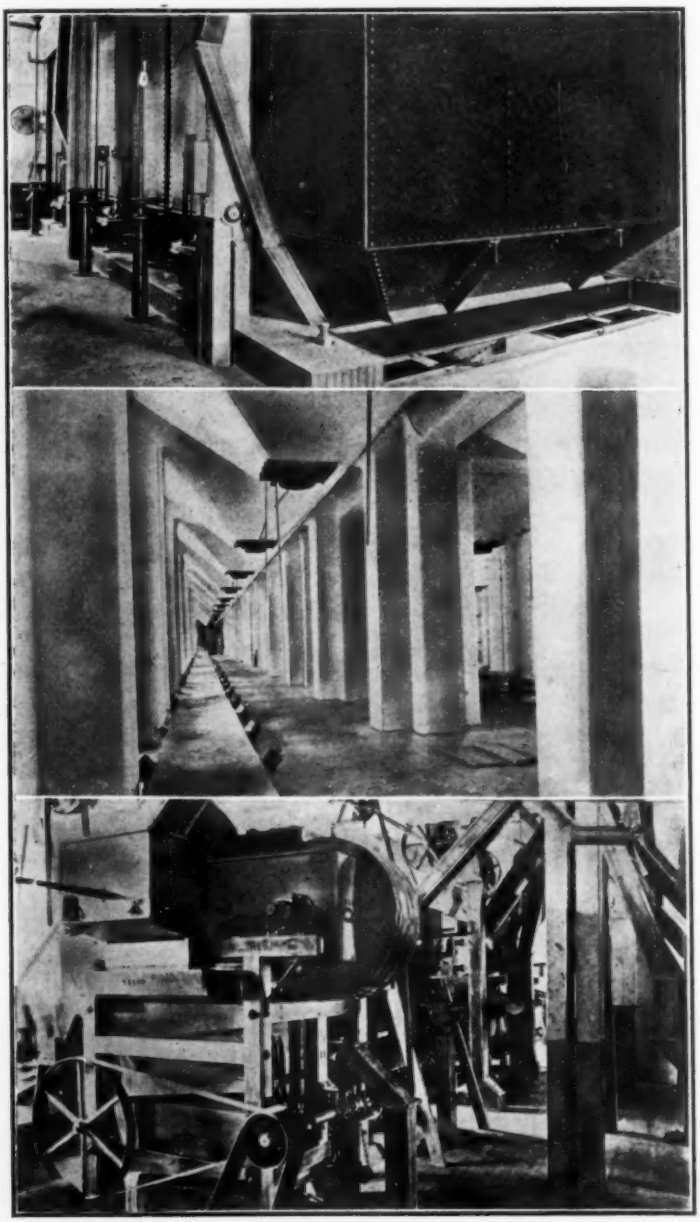

FIG. 2 THREE HOPPER SCALES OF 2000 BUSHELS CAPACITY EACH

FIG. 3 THE FIRST FLOOR OF THE ELEVATOR SHOWING THE CONCRETE CONSTRUCTION FOR THE SUPPORT OF THE BINS; THE BIN HOPPERS; THE BELT CONVEYORS FOR MOVING THE WHEAT FROM THE BINS TO THE WHEAT CLEANING HOUSE. SIMILAR BELTS IN THE ATTIC DISTRIBUTE THE WHEAT INTO THE BINS

FIG. 4 MILLING SEPARATORS FOR REMOVING SEEDS AND FOREIGN SUBSTANCES FROM THE WHEAT

result of these experiments, was not attempted until 1878 or 1879 when C. C. Washburn built an experimental mill using corrugated and porcelain rolls exclusive of stone. A complete roller mill had been built in Presth, Hungary, and in 1864 an engineer from this mill built one at Fiume, Austria.

16 The old hexagon reels for bolting flour which had been in use in the old style mills were very clumsy pieces of machinery. They were built from 16 to 20 ft. long, but were gradually replaced by shorter round reels and centrifugal reels, about 32 in. in diameter, 8 to 10 ft. long with beaters revolving in the direction of the reel at a higher speed than the cylinder. This was an important improvement. The centrifugal reel was of European origin, but it was improved in the manner of drives and the construction made lighter and simpler. A system of bolting the flour and grading the middlings had been experimented with, requiring less space with the added advantage of having larger capacity.

17 In 1892 a sieve machine was installed in a Minneapolis mill by Carl Haggenmacher that was destined to revolutionize the bolting system of milling. It is a long call from the sifting of the meal by hand as shown in the hieroglyphics of Egypt, and yet these later improved bolting sifters work on the ancient principle of the hand sieve.

18 The plansifter consists of a box with a nest of sieves alternating with blank carriers to take off the different separations. The box is suspended by hangers and a gyrating motion imparted to it by an eccentric in a counterbalanced driving pulley on a vertical shaft attached to the underside of the sieve frame. The introduction of this sifter disabused the minds of many millers of the idea that returns or cut-offs were essential.

19 Next was developed the square sifter, a machine made of four separate square boxes containing a nest of sieves about 20 in. square, placed on top of each other and arranged in series for different separations. The product tails from the upper sieve to the lower or to the discharge spout, the finished product or siftings being gathered under each sieve on blanks and spouted out of the machine through openings in the sides of the frame. The motion of the square sifter as in the plansifter is gyratory, the principle of using the small sieves placed vertically over each other being identical with the use of four short reels 2 ft. long instead of one reel 8 ft. long. The cut-off system is provided for. In the different forms and makes of sifters, the great advantage lies in the floor space required for the

bolting capacity over the reels, which is especially true in the vertically constructed sifters.

20 In 1895 the universal bolter, a sifter of novel construction of the vertical type, was patented and gradually developed first as driven by a crankshaft running in rigid boxes fixed on the top and bottom heads of the frame. The upper end of the shaft ran in a stationary bearing and the bottom in a step set rigid. The frame was counterbalanced at the top and bottom of the shaft by heavy wheels, and the gyrating frame was suspended by rods. The objection to this fixed eccentric was the difficulty of balancing the machine under a varying load which conveyed a vibration to the building. To overcome this was applied a novel patent drive described by the patentee W. Worby Beaumont, as "dispensing entirely with the crank shaft and using rotary balance weights fixed to a straight vertical shaft running in bearings attached to the frame or body to be gyrated. The crank and connecting rod being absent, the balance weights are unbalanced except dynamically and the want of balance is kinetically equivalent to the required motion of the body to be gyrated."

21 To adapt this movement to practical use improvements in the drive connections and a device for restricting the abnormal outward throw of the bolter at starting or stopping or when not running at the critical speed of rotation, have been made and patented.

22 By referring to Fig. 21 the shaft or spindle A is carried in bearings attached to the frame B to which it is desired to give a gyratory motion. The frame is freely suspended and the shaft and frame attached are restricted to a circle proportional to the inertia of the frame and of the mass of the weight C considered with reference to the distance of its center of mass from the center of the shaft A. Shaft A is driven through a flexible or universal coupling by a driving shaft in a fixed bearing. The unbalanced eccentric weight C is carried by an arm secured to shaft A. By changing the radius of the weight or adding weight, the radius of the gyratory movement of the frame B is increased. A wide range of speed or in the weight of the frame B may be used without affecting the diameter of the gyrating circle described by the frame, which is unrestrained except by the flexible suspension rods D. When in motion considerable pressure would be required to throw it out of its gyrating circle. The pins E at the top and bottom of the shaft are adjusted to half the throw of the frame; centers plumb and steady when running at the critical speed of rotation. This movement is

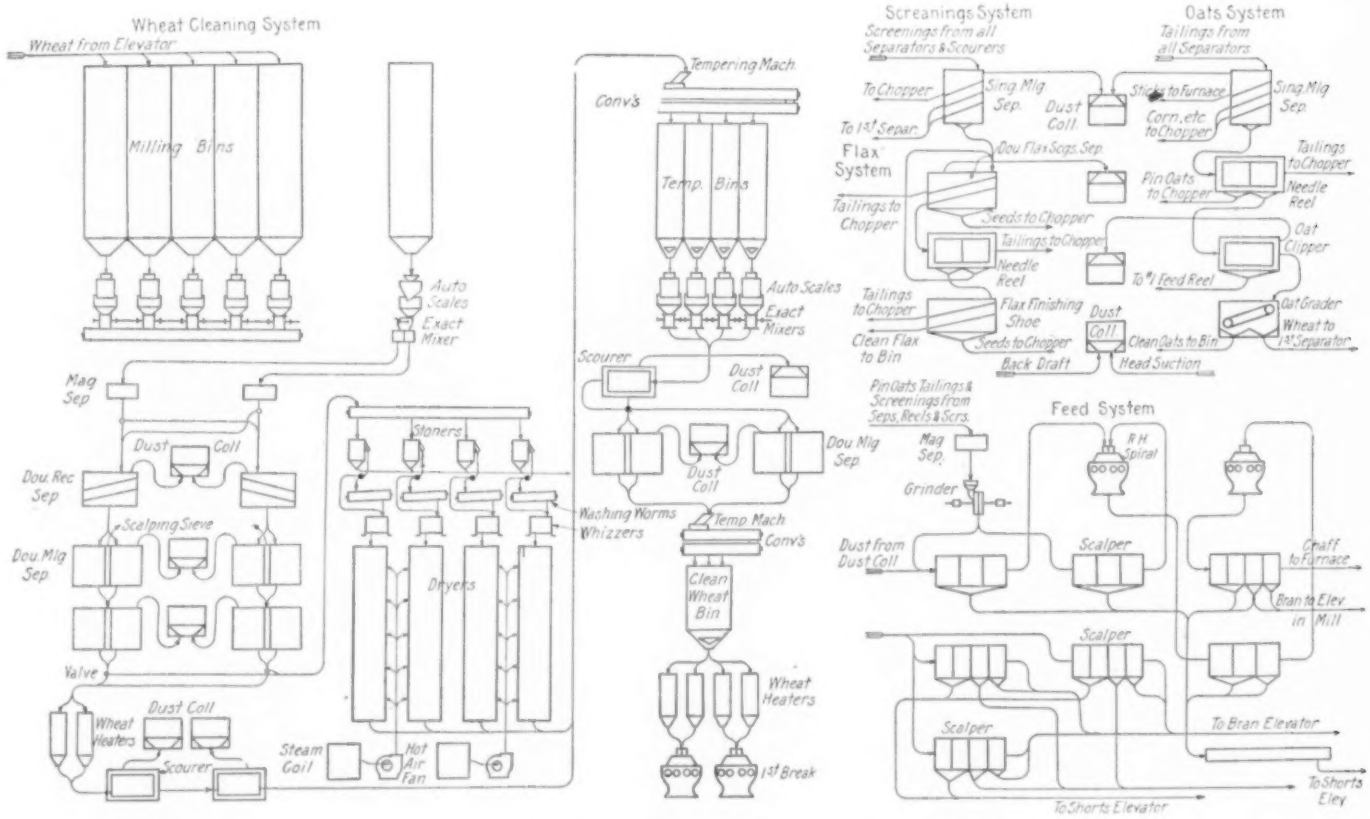

FIG. 5 DIAGRAM OF THE PROCESS OF CLEANING THE WHEAT

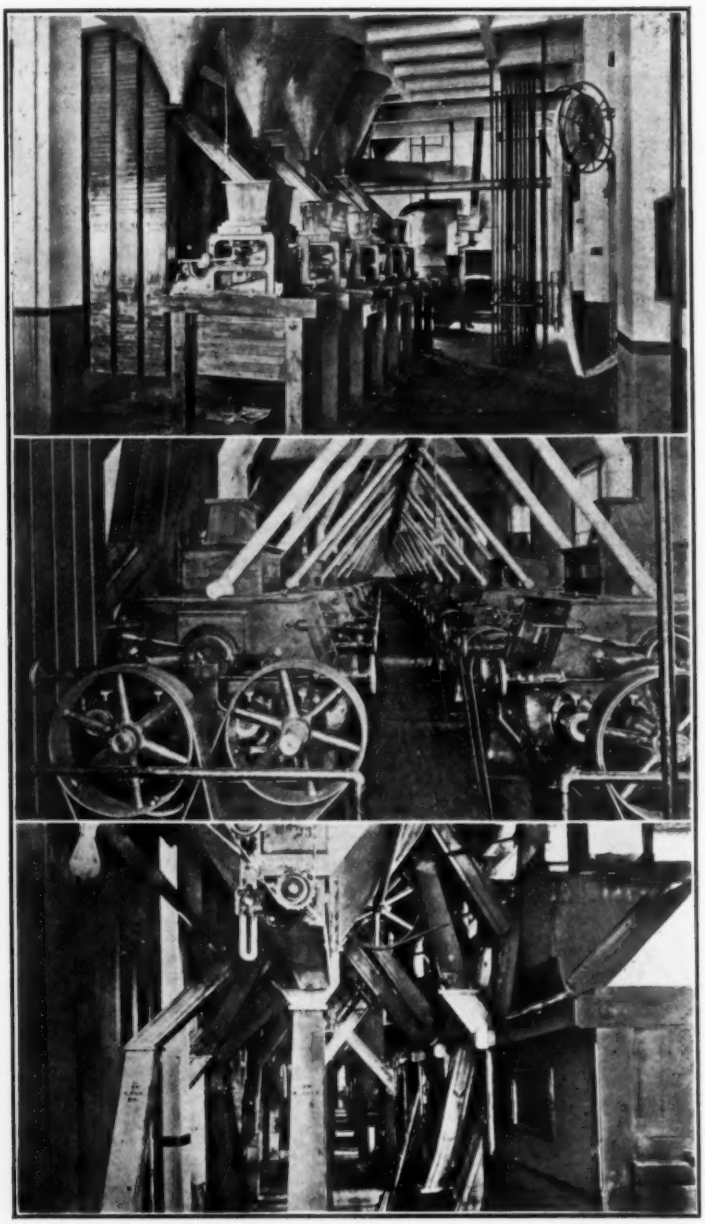

FIG. 6 AUTOMATIC SCALES FOR WEIGHING THE WHEAT FROM THE ELEVATOR INTO THE MILL
FIG. 7 VIEW OF THE ROLLER MILLS ON THE SECOND FLOOR
FIG. 8 VIEW OF THE PURIFIERS AND PARTIAL VIEW OF THE DUST COLLECTORS ON THE FIFTH FLOOR

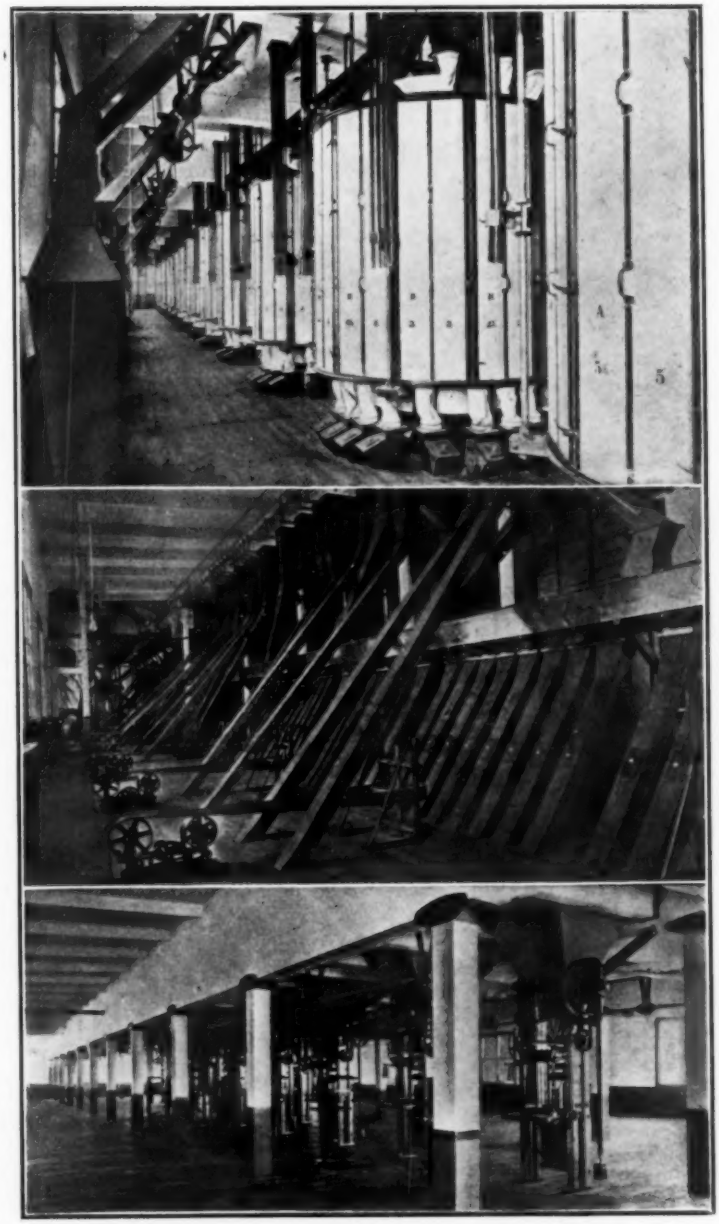

FIG. 9 VIEW OF THE SIXTH FLOOR SHOWING THE UNIVERSAL BOLTERS

FIG. 10 VIEW OF THE ATTIC, OR SEVENTH FLOOR, SHOWING THE ARRANGEMENT OF SPOUTS FROM THE ELEVATORS TO THE UNIVERSAL BOLTERS AND THE BELT DRIVE FOR THE UNIVERSAL BOLTERS

FIG. 11 VIEW OF THE PACKING FLOOR SHOWING THE FLOUR PACKERS IN THE FOURTH STORY OF THE WAREHOUSE

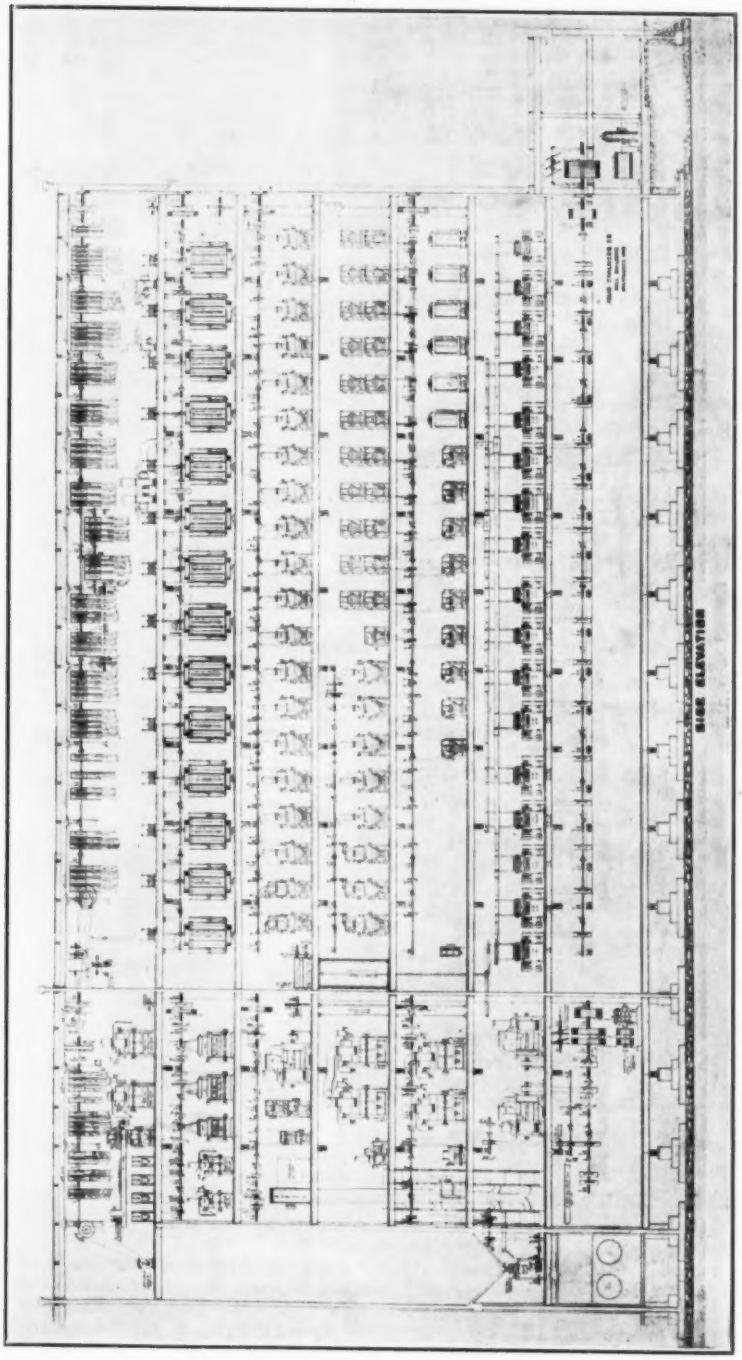

FIG. 12 LONGITUDINAL SECTION OF THE MILL, MOTOR-DRIVE HOUSE AND WHEAT CLEANING HOUSE

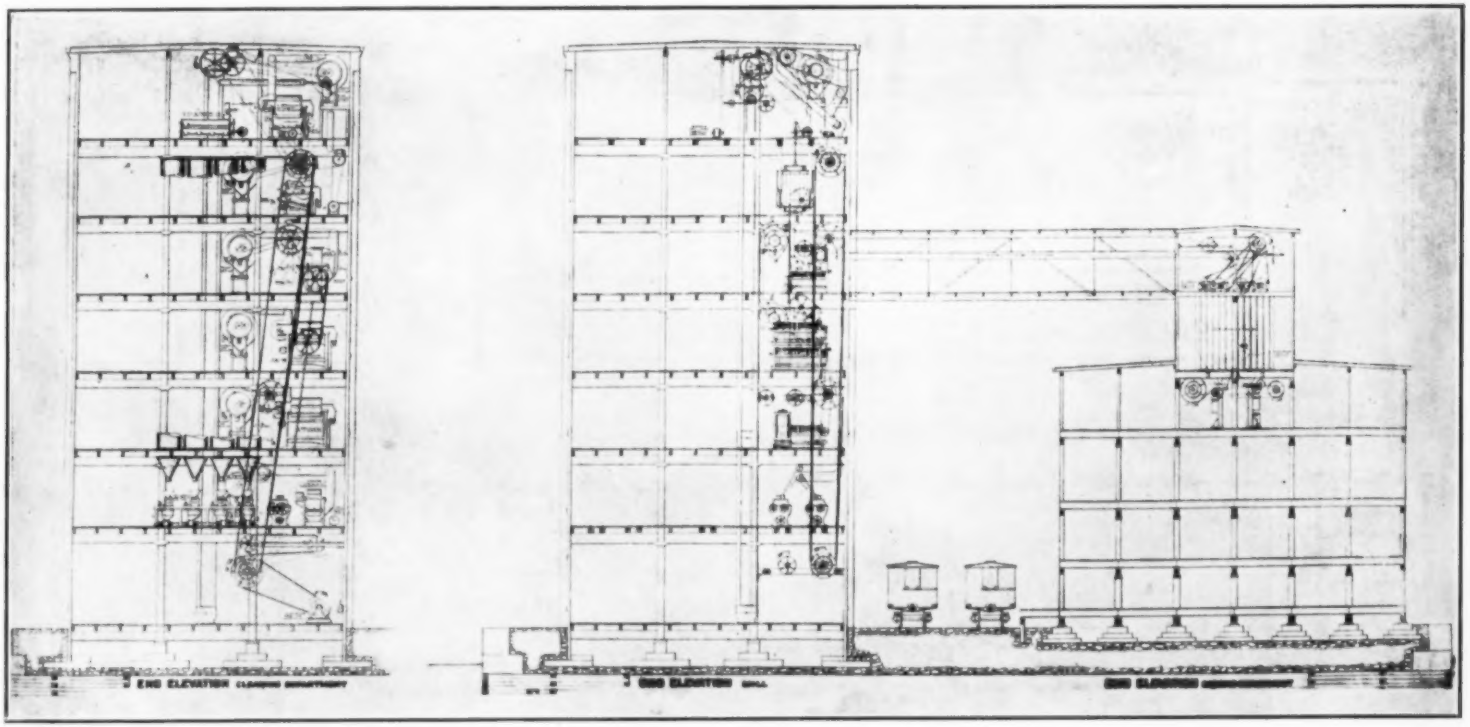

FIG. 13 SECTIONAL DRAWING OF THE END ELEVATION OF THE MILL SHOWING PACKING ROOM IN THE FOURTH STORY OF THE WAREHOUSE, ALSO SECTION OF THE END ELEVATION OF THE WHEAT CLEANING ROOM

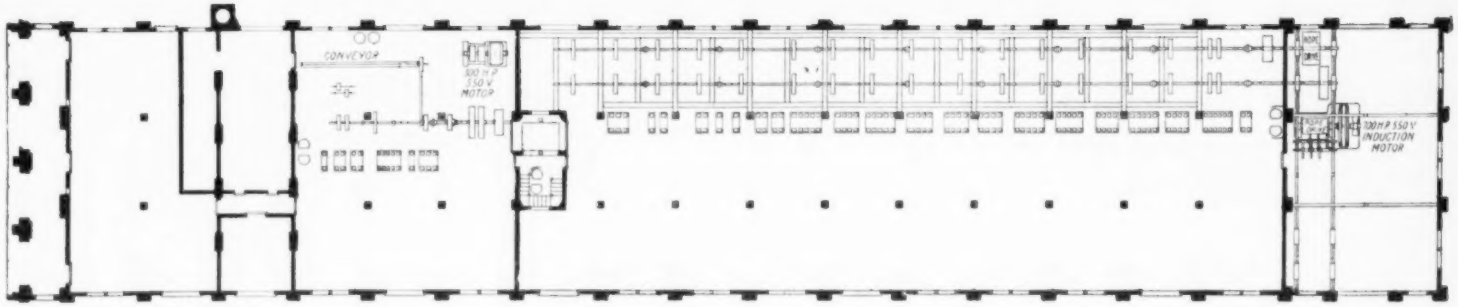

Fig. 14 First Story Detail Drawings of the Mill and Cleaning House

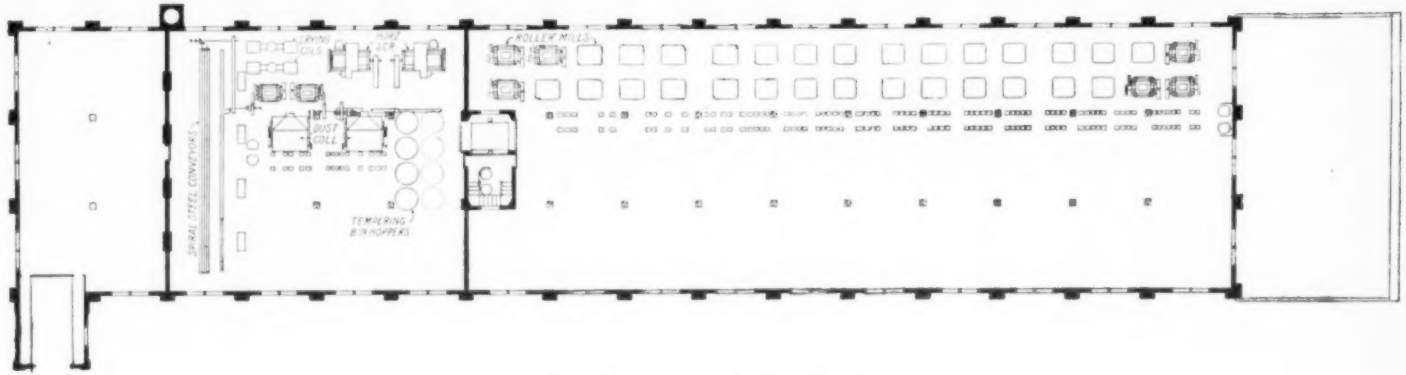

Fig. 15 Second Floor Plan

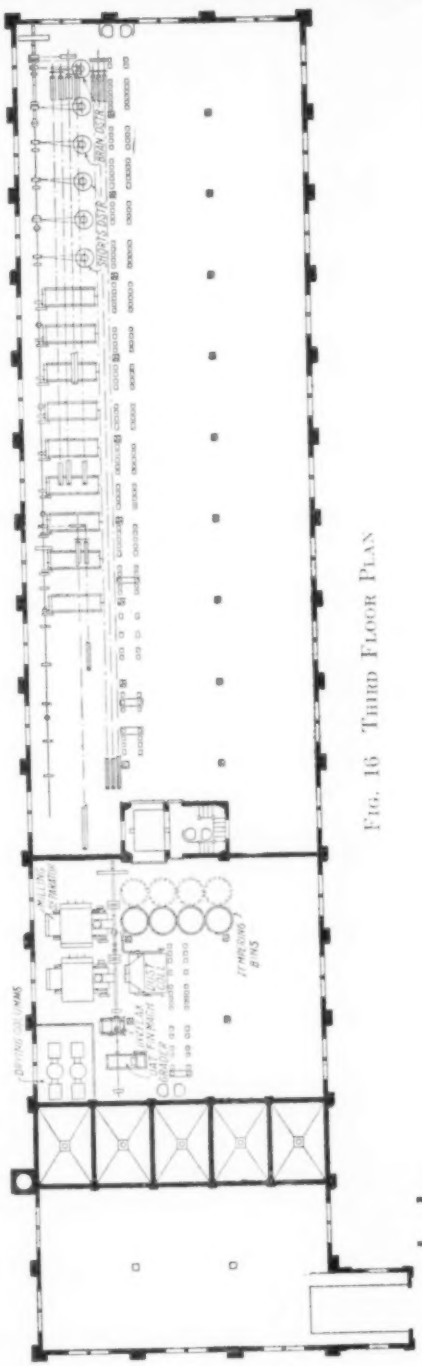

FIG. 16 THIRD FLOOR PLAN

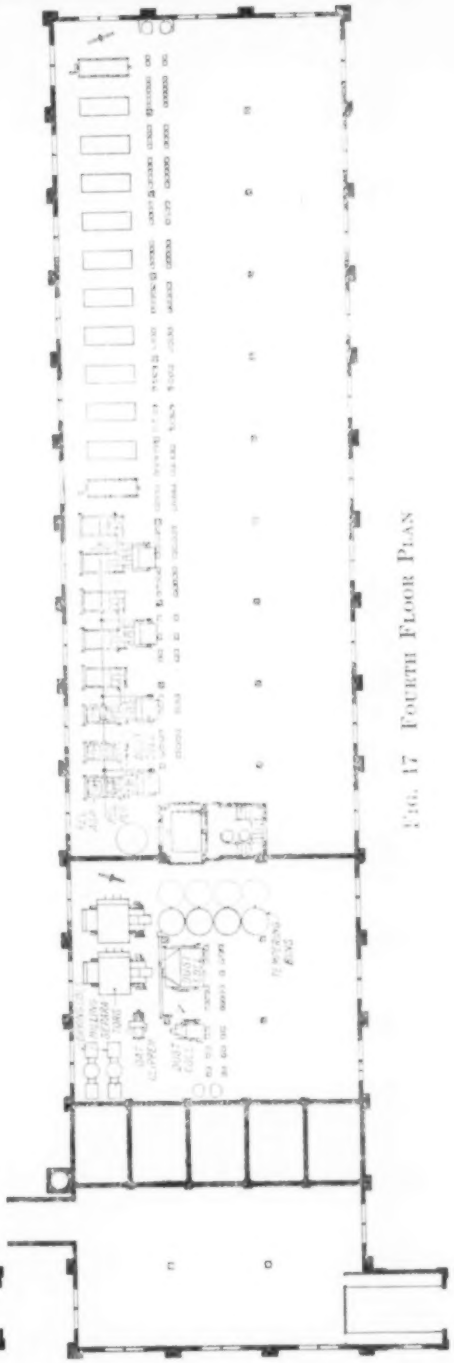

FIG. 17 FOURTH FLOOR PLAN

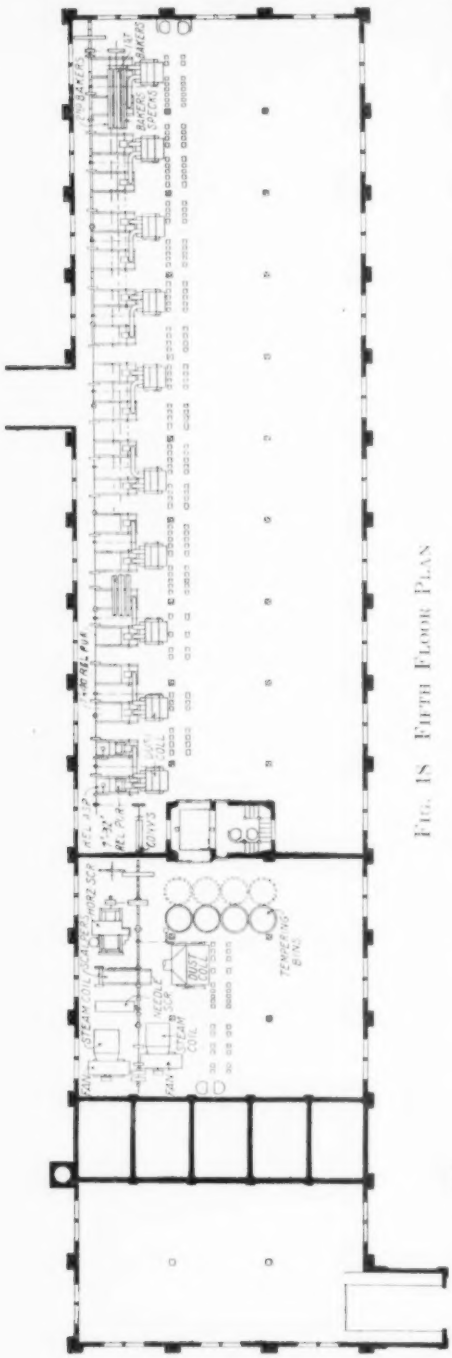

FIG. 18 FIFTH FLOOR PLAN

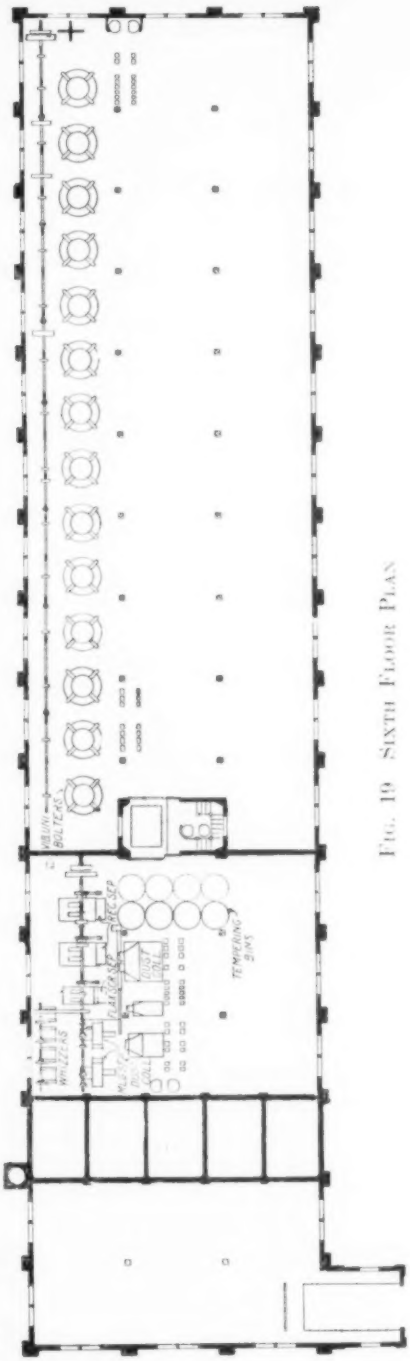

FIG. 19 SIXTH FLOOR PLAN

FIG. 20 SEVENTH OR ATTIC FLOOR PLAN

called a vibromotor drive because a large part of the motion imparted by it is the result of converting into useful work the waste of eccentrics.

23 The arrangement of sieves in the universal bolter is a novel system. The frame is dodecagon and the conical sieves are supported in grooved slides, each sieve having a drop of $\frac{1}{2}$ in. and set level in the frame. The $\frac{1}{2}$ in. drop in each sieve continues around the 12 sides, forming a series of descending spirals and affording a continuous bolting surface proportioned to the requirement. The material is admitted to the head sieve by spouts in the outside doors and discharged where necessary through similar spouts. The principle of gyrating sieve bolting is that the stock is not continually tossed and mixed as in a reel, but is sifted through the silk, the fluff being floated on the top of the stock over the tail leaving a brighter and purer flour.

24 An important factor in a flour mill is the cleaning machinery, consisting of separators for removing chaff, straw, foreign seeds, etc., and scourers for polishing the wheat and removing the dust. Washers and dryers are also essential appurtenances.

25 The tempering of the wheat requires the greatest attention and skill as the varieties grown in the different sections need different treatment to obtain the best results. The analysis and chemistry of wheat are being given much attention by most of the state agricultural colleges with the object of improving the quality and selecting the varieties of wheat best adapted for milling purposes. Science is being applied to the art of milling and the utmost care is used in selecting a mixture that insures the requisite qualities of a good bread-making flour. There is no food manufactured that is purer or so entirely free from adulterations.

26 This brief history of the development in the art of milling in America will suffice to introduce a description of a modern flour mill built on a system which has been evolved step by step. It is the product of many minds; many features are peculiarly American, especially the labor saving devices.

27 Within the last few years the tendency has been to build mills of large capacities ranging from 2000 to 6000 bbl. every 24 hours. For such mills the merchant miller chooses a location having advantageous traffic rates, cheap power and near to the market. Some representative mills of this class are the mill in New York City of 10,000 bbl. capacity in two units; the 13,000 bbl. mill in three units (one 7000 and two 3000 bbl. each) in Buffalo; 5000 bbl. in one unit

in Keewatin, Ontario; 2500 bbl. in one unit in Montreal; the 4000 bbl. in one unit at Winona, Minn., now building; the Maple Leaf Milling Company of 4000 bbl. capacity in one unit at Port Colborne, Ontario. The last two mills are to have an ultimate capacity of 9000 bbl. each.

28 These mills are water, steam and electrically driven and represent the latest and most advanced types both in mechanical construction and in the art of milling. The Maple Leaf Mill at Port Colborne has been selected for the subject of illustration as it is the latest complete mill in operation. It is electrically driven, Fig. 1 showing the power station which contains three 800-kw. transformers taking power from Niagara at 23,000 volts and transforming to 550 volts.

29 The front view of the mill (Fig. 1) shows the motor house, the mill building 60 ft. by 264 ft. and seven stories high. The storage elevator of 60 bins of 15,000 bushels each, with the storage capacity in the mill makes the present wheat storage capacity 1,000,000 bushels. On the installation of the second unit 40 additional bins are to be added, making a total capacity of 1,600,000 bushels. The marine leg shown in elevation has a capacity of 28,000 bushels per hour. The mill is located at the entrance of the Welland Canal, and boats of 20 ft. draft carrying 350,000 bushels of wheat can be loaded from the great elevators at Fort William or Duluth at the head of Lake Superior and discharge their cargo without breaking bulk directly into the mill elevator. At the close of navigation the boats are used for the storage of their last shipment, until the opening of navigation, thus greatly increasing the storage capacity. The elevator is entirely of reinforced concrete construction; the mill building walls are of a similar material and the interior is of timber.

30 The three hopper scales each of 2000 bushels capacity, in which the wheat from the marine leg is weighed, are shown in Fig. 2. Fig. 3 is a view in the elevator showing the two traveling belts for moving the wheat from the bins to the wheat cleaning house. On the top floor are similar belts from which the wheat is spouted to the different bins.

31 A diagram of the process of cleaning the wheat is shown in Fig. 5. The wheat after passing over receiving separators is delivered to the storage bins in the wheat cleaning house, after which it is weighed on automatic scales and passed over a magnetic separator. It is then spouted over sieve separators to remove the foreign

matter and from these receiving separators over the milling separators. There are four of these milling separators, the stream being split as shown in the diagram, thus affording a double separation. From these milling separators the clean wheat passes through heaters or tempering machines, the object being to toughen the outside coating of the wheat or bran. It then goes through the process of scouring to remove the fluff or beard on the wheat, this action producing a polish on the grain. From these scourers the wheat is spouted to the wetting conveyors. A certain amount of water is applied to the wheat with the object of properly tempering and conditioning the grain. From the tempering bins it is weighed over automatic scales and the difference between the gross weight and this net weight gives the exact amount of seeds and dust or loss between the wheat from the elevator and the clean wheat going to the first reduction in the mill. After being weighed on these automatic scales it is given another scouring and passed over another milling separator to remove what dust has accumulated in the process. Tempering conveyors are used before spouting the product to the clean wheat bins. From these bins it passes through a set of wheat heaters in order to keep the tempering uniform just before being spouted to the first break roller mills in the grinding process.

32 In case the wheat is smutty or dirty it is switched after leaving the first four milling separators, as shown by the valve in the diagram, to stoners and washing machines. After being thoroughly washed it is passed through whizzers which separate the water from it. The wheat is then passed through dryers in which both hot and cold air is used. The process after leaving the dryers is the same as in the case of the wheat not washed. Figs. 4 and 8 show sections of the wheat cleaning department.

33 There are other systems of screening machines used for separating the flax, oats and mustard seeds.

34 Fig. 12 is a longitudinal section of the mill, motor-drive house and wheat cleaning house. The first floor shows the two lineshafts for driving the double line of roller mills; the second or roller floor shows 14 double pair of 10 by 42 and 20 double pair of 10 by 36 roller mills (Fig. 7); the third floor has a few reels, six bran and shorts dusters, conveyors for gathering flour from the different bolters, and spouting to the rolls and elevators; the fourth floor contains 23 centrifugal reels for dressing the lower grades of flour and for blending and mixing purposes; the fifth floor contains the middlings purifiers (Fig. 8), of which there are 28 and dust collectors

for every two purifiers; the sixth floor is the bolting floor containing 14 vibromotor universal bolters (Fig. 9) of the largest capacity; the seventh floor is the attic showing the elevator lineshafts and the drives for universal bolters.

35 The universal bolters will take care of a stream on the breaks for a 5000 bbl. mill. The main stream is divided into eight parts. The first separation is over a 24 wire, the tail of which is spouted through spouts to the fifth break rolls; the sifts of the 24 wire pass over 32 wire to the coarse middlings purifiers; the stock sifted through the 32 wire is bolted over a 46 special wire and the tails spouted to the first middlings purifiers; the sifts of the 46 wire are bolted on to 58 special wire and the tail spouted to the second middlings purifiers; the sifts of the 58 special wire are dropped to a 72 special wire and tailed to the third middlings purifier; the sifts of the 72 are bolted on No. 9 silk, the tail going to the fourth middlings reduction and the siftings to the flour dresser. Thus it will be noted that six separations are made on this machine. The spiral arrangement of the sieves enables the stock to flow continuously over the wire or silk until the separation is completed and the amount of wire may be proportioned to the amount of dusting or scalping required. The single machine has a capacity of ten centrifugal reels and occupies a floor space of 36 sq. ft. All the space required for the bolting capacity of a 4000 bbl. mill as shown in Fig. 19 is 4920 sq. ft.

36 A view of the attic or seventh floor of the mill, the arrangement of spouts from the elevators to the universal bolters and the belt drive for the universal bolters is shown in Fig. 10.

38 Fig. 13 is a sectional drawing of the end elevation of the mill and packing room (see also Fig. 11); Figs. 14 to 20 are detail drawings of the various floor plans of the mill.

39 A 700-h.p. induction motor of the wound rotor type, running at a speed of 485 r.p.m. drives the two lineshafts. It is supplied with current at 550 volts and $66\frac{2}{3}$ cycles. The wound rotor type is used in order to obtain a high starting torque, necessary for flour-mill work, with moderate live current.

40 The sheave on the motorshaft is 40 in. in diameter, 36 grooves for $1\frac{1}{4}$-in. rope divided into three sections of 12 ropes, each section having a separate take-up; the first 12 ropes drive one roller lineshaft, the remaining 24 drive the other lineshaft for which the drive for the upper part of the mill is taken off by a rope sheave 64 in. in diameter, 9 grooves, $1\frac{3}{4}$-in. rope.

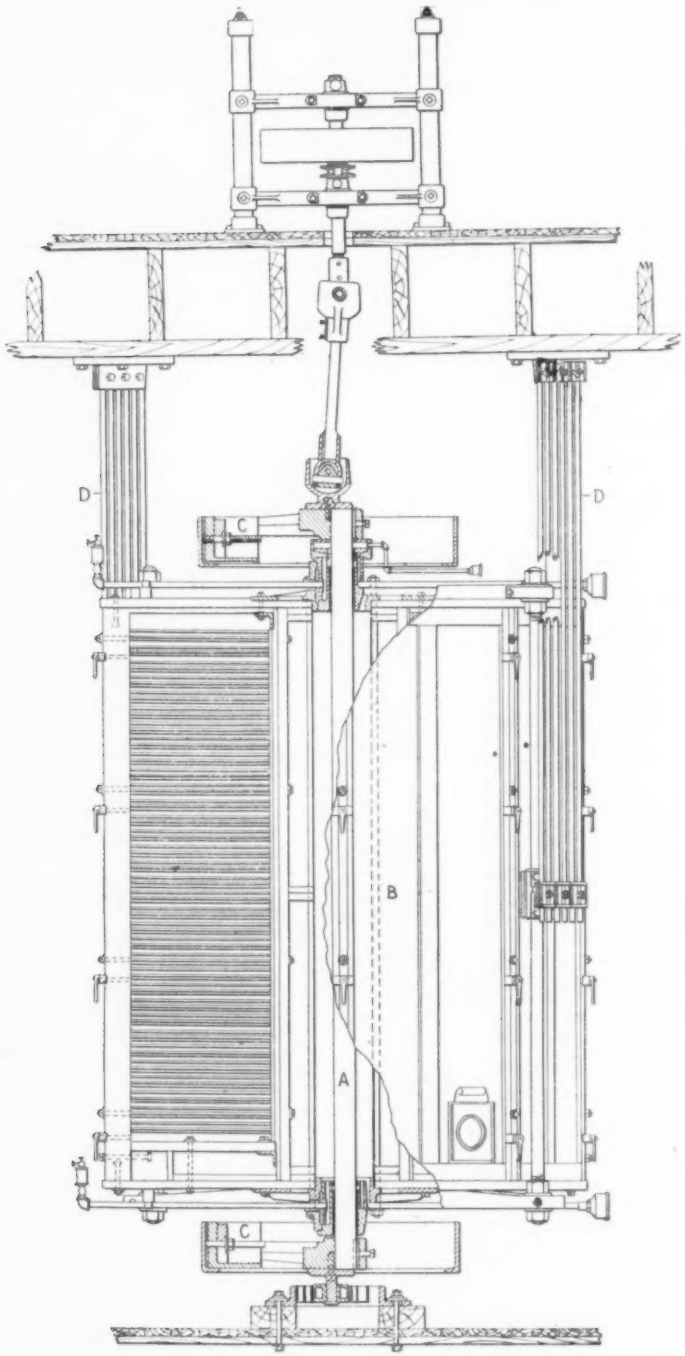

FIG. 21 UNIVERSAL BOLTER WITH NOVEL ARRANGEMENT FOR PRODUCING GYRATORY MOTION OF FRAME

41 The cleaning machinery is driven by a 300-h.p. wound rotor motor running at 485 r.p.m., the sheaves on the motorshaft being 38 in. in diameter, 16 grooves, $1\frac{1}{4}$-in. rope driving an 80-in. sheave on the lineshaft. The upstairs machinery is driven from this lineshaft by a 64-in. sheave, 8 grooves, $1\frac{3}{4}$-in. rope. The marine leg, the wheat receiving machinery in the elevator and the flour packers are all driven by separate motors.

42 The engineering and mechanical construction of a flour mill is carried out with the object of incorporating a system of milling to produce results in the manufacture of flour as regards quality, yield and percentages of the different grades.

43 The majority of American mills make seven grades of flour, first and second patent, first and second clear commonly called baker's flour, low grade and red dog; or by blending the four first grades making a straight grade of flour. The percentages and yield vary according to the quality of the wheat.

44 Using a good grade of wheat and making a yield of 4 bushels and 16 lb. to a bbl. (196 lb. of flour) it is possible to obtain $61\frac{1}{2}$ per cent patent flour, 9.2 per cent of baker's flour, 3.7 per cent of low grade, 2.4 per cent of red dog, leaving 23.5 per cent of offal which includes bran and shorts. The quality of the two low grades of flour of course would be governed by the yield and the closer the yield, the poorer those grades of flour would be.

45 The clean wheat from the tempering bins is delivered to the first two stands of 10 by 42 roller mills. These are corrugated or grooved with special patterns (12 to the inch), and 168 in. of surface is contained in these four pairs of rolls running at 500 r.p.m. with a differential of $2\frac{1}{2}$ to 1. This particular system is called the first break rolls, five breaks being used in this mill to reduce the wheat to bran. Figured on a yield of 4 bushels and 16 lb. per barrel of all grades of flour, 702 bushels of wheat per hour pass through these first break roller mills. On the first break the wheat is broken down so as to disintegrate the inside of the kernel into granules called semolina or more commonly termed middlings. It is scalped on a universal bolter and the tailings that pass over 16 wire tail to the second reduction on two double stands of 10 by 42 roller mills corrugated 14 to the inch and scalped over 18 wire; the tail of the scalping of the second break passes to the third break roll corrugated 16 to the inch and scalped over 20 wire; the tail of the scalping of the third break passes to the fourth break rolls corrugated 20 to the inch and scalped over 24 wire; the tail going to the fifth break rolls

corrugated 24 to the inch. The fifth break is scalped on reels, clothed with 30 wire, the tail of these reels being bran and dusted on bran dusters. On each of these breaks there are six different separations of middlings taken off and the residue being flour is dressed separately.

46 Of the five grades of middlings thus separated, the coarse scalped over 28 wire contain and are mingled with small particles of bran and the germ of the wheat, each separation being finer until the last separation scalped over a No. 9 silk contains very fine particles of bran. Each of these grades of middlings is sent to separate purifiers.

47 The operation of the purifiers is to remove the fluff and dust and tail over the coarser or branny particles called tailings. The siftings of the purifiers are reduced on smooth rolls by a gradual reduction; the coarse middlings are reduced so as to disintegrate the bran particles, the starch cells and germ, the semolina being again reduced to flour; the tailings are repurified and further reduced on smooth rolls and treated by several reductions as with the coarse middlings.

48 In all there are 19 reductions on smooth roller mills, the separations required to eliminate the germ, linings of the starch cells, fine bran called shorts and the bran, being very elaborate and necessitating exact judgment in selecting the proper numbers of wire or silk and skill in the art of grinding to accomplish the proper results.

49 The fine silk used for bolting the flour is manufactured almost exclusively in Switzerland by weavers on hand looms in their own homes. Great perfection in the evenness of the meshes is obtained, as fine as 200 meshes to the inch being made.

50 It is not possible to give in this paper more than a general outline of the process of the art of milling by gradual reduction, but the progress in the mechanical construction of a modern flour mill, which has by no means reached perfection either mechanically or in the art, can only be illustrated. Continual improvements are being made and one idea brings out another.

A NEW ANALYSIS OF THE CYLINDER PERFORMANCE OF RECIPROCATING ENGINES

By J. Paul Clayton

A PREFATORY STATEMENT

Early students in the field of thermodynamics gave generous attention to a study of the steam-engine indicator diagram. Attempts to deduce a law which would connect the form of the expansion curve of such diagrams with other factors affecting engine performance have, however, always failed, and in recent years studies involving the precise form of such curves have appeared unprofitable. It has remained for Mr. Clayton to discover the key which reveals the hidden relationships. The methods he has employed are so simple that they may be readily understood even by the casual reader. His conclusions are well sustained and are of sufficient significance to advance by a long step forward our understanding of the steam-engine problem.

Mr. Clayton's researches have been made under the auspices of the Engineering Experiment Station of the University of Illinois, and the results which he presents herewith will in due time be published in more complete form as a bulletin of the station.

W. F. M. Goss

ABSTRACT OF PAPER

Our knowledge of the cylinder performance of reciprocating engines is obtained almost entirely from indicator diagrams. These diagrams provide a measure of the work performed, thus enabling the efficiency or economy of the engine to be determined; they also provide an aid for setting valves, and furnish the basis in steam cylinders for applying Hirn's analysis, for measuring the initial condensation, and for finding the diagram factor for the purposes of design. It has been generally thought by engineers that the evidence contained in the diagram was limited only to these and minor uses.

The investigation described in this paper has disclosed the fact that the indicator diagram contains in itself the evidence necessary for an almost complete analysis of cylinder performance, the results of which have not heretofore been considered possible.

In obtaining these results the diagram has been transferred to logarithmic cross-section paper and thus a figure has been drawn which will be called a logarithmic diagram. By the aid of this diagram it has been found that the expansion and compression curves of all elastic media used in practise obey substantially the polytropic law $PV^n = C$. From this fact there have been developed rational methods of approximating the clearance of a cylinder, of closely

locating the cyclic events, and of detecting moderate leakage with the engine in normal operation.

It has been discovered that the value of n in the law $PV^n = C$ is controlled directly in steam cylinders by the value of the quality of the steam mixture at cut-off, called X_c, and that the relation of X_c and n is practically independent of cylinder size and of engine speed for the same class of engine. This fact enables us to determine the actual amount of steam and water present in a cylinder at cut-off from the experimentally determined relations of X_c and n, and thus to obtain the actual steam consumed from the diagram.

The paper contains an exposition of the graphical methods employed, charts giving the relations of X_c and n for determining the steam consumption from the diagram, and examples of typical logarithmic diagrams.

A NEW ANALYSIS OF THE CYLINDER PERFORMANCE OF RECIPROCATING ENGINES

By J. Paul Clayton, Urbana, Ill.

Junior Member of the Society

The investigation described in this paper is the result of an extensive analytical and experimental study of the forms of the expansion and compression curves which occur in indicator diagrams. From this study has been developed a new and very complete analysis of cylinder performance for reciprocating engines using any elastic fluid for the working medium.

2 The analytical study was carried on by transferring the indicator diagram to logarithmic cross-section paper and thus drawing a figure which will be called a logarithmic diagram. It is well known that the equation of the polytropic curve $PV^n = C$ becomes a straight line when plotted on logarithmic cross-section paper. Conversely, when the expansion or compression curve of an indicator diagram becomes a straight line in the logarithmic diagram, then the curve is of the form $PV^n = C$, the value of n being the slope of the line.

3 The logarithmic diagram is more useful for analysis than any other form of diagram because of the natural limitations of the human mind. We do not possess the power to distinguish between curves, but are able, however, to see clearly the differences in these curves after they have been transformed into straight lines, which fact alone makes these new methods of analysis possible. We are now enabled in their straight-line form to comprehend curves which we have always seen, but could not distinguish one from the other in their original form.

4 By means of the logarithmic diagram it has been found that, free from certain abnormal influences, expansion or compression of an elastic medium takes place in the cylinders of reciprocating engines substantially according to the law $PV^n = C$.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

5 From the fact that the law $PV^n = C$ holds for expansion and compression curves from practice, there have been developed rational methods of approximating the clearance volume, of closely locating the cyclic events, and of detecting moderate leakage when the engine is in regular operation. These methods apply, however, only to those indicator diagrams which are taken from the cylinders of reciprocating engines using any elastic fluid for the working medium and having as a part of the cycle an expansion or compression of the medium.

6 It has been discovered that the value of n for the expansion curves of steam diagrams bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off. This proportion or quality will be called X_c in this investigation, and its value will be expressed in decimal parts of unity. The relation of the value of n to the value of X_c for the same class of cylinder as regards jacketing has been found to be practically independent of engine speed and of cylinder size.

7 The practical significance of finding this relation is that there is now available an accurate method of approximating the value of X_c and therefore the actual weight of steam and water present at cut-off from the indicator diagram alone.

8 It is believed, after working with the new methods of analysis, that the existing methods are in a very crude state, and that many of the deductions made from them are without foundation or meaning.

9 Preliminary to the investigation, an examination was made of the nature and form of the expansion and compression curves from a large number of steam engines, by the aid of the logarithmic diagram.

10 Values of n in the equation expressing the law $PV^n = C$, however, exhibited a large range of variation, the range being from 0.70 to 1.34. The engines from which the values were obtained differed in type, size, speed, steam pressure, ratio of expansion and back pressure. Obviously, comparisons could not be made of these examples because of the number and magnitude of the variables.

11 Indicator diagrams taken from the same cylinder with different cut-off positions showed that the value of n was higher as the cut-off was lengthened. There was a large variation in the value of n where the conditions of cylinder size, speed, steam pressure and steam distribution were the same. The only variable shown by the diagrams was length of the cut-off. After considerable study, it was

decided that there might be some relation in any one cylinder between the value of n for the expansion curve and the value of X_e, as the value of X_e was known to be higher as the length of cut-off increased. One fact that seemed to confirm this hypothesis was that, with superheated steam (under the same general conditions, except the kind of steam used), when the value of X_e is high, the value of n for expansion is always much higher than with saturated steam. The only important variable between the use of saturated and superheated steam to account for the change in the value of n was the quality of the steam mixture at cut-off, or the value of X_e.

12 All cases of engines examined using superheated steam at normal cut-off showed n to be higher than 1.0, and as high as 1.34; and all cases of small engines using saturated steam showed n to be lower than 1.0 and as low as 0.70. These facts led to the conclusion that the value of X_e was the most important single factor in the accompanying value of n. Tests were therefore planned in which the effort was made to vary the value of X_e between the widest practicable limits.

LABORATORY TESTS

13 An outline of these tests will first be given, with a synopsis of the method of analysis used, after which the application of the analysis to indicator diagrams will be given for the purpose of determining cylinder performance.

14 A single-cylinder, long-range cut-off, 12-in. by 24-in., Corliss engine, located in the mechanical engineering laboratory of the University of Illinois, was selected for the tests. A Corliss engine was selected because of the fact that in this type all the steam used passes through the cylinder.

15 It was planned to observe the effect upon the value of n of varying the value of X_e under different conditions of pressure and speed; n being the exponent in the equation $PV^n=C$ for the expansion curve of an indicator diagram, and X_e the quality of the steam at cut-off. The value of X_e was varied through a large range by the use of saturated and superheated steam, in conjunction with different lengths of cut-off under the same conditions of pressure and speed. The values of X_e obtained ranged from 0.50 to 0.90, covering the range usually found in practice with the type of engine used.

16 The values of n for the expansion curves were obtained by means of the logarithmic diagram explained in detail in the Appendix. The value of X_e given in the log is the average of the results

obtained from one set of head-end and crank-end diagrams for each test. The unit of measurement, therefore, was the revolution, as the values of X_e for the head and crank ends cannot be measured separately when one exhaust pipe is used for both ends of the cylinder. The value of n given for one test is the average of the separate values from the expansion curves of the head-end and crank-end diagrams, taken from the set of diagrams already mentioned.

17 Seventy-four tests in 16 series were run. Of this number, 60 tests in 14 series were selected as fulfilling the requirements decided upon to give reliable data. Each series consisted of four to five separate tests, differing from each other only in the length of cut-off, with the same conditions of pressure and speed. All tests were run with the steam exhausting from the cylinder at about atmospheric pressure into a surface condenser. The length of cut-off was varied in nearly uniform steps from about 5 per cent to 45 per cent of the length of the stroke, and was the means of varying the value of X_e when using either saturated or superheated steam.

18 The 14 series were divided into two divisions of seven series each; one division being run with saturated steam, and the other with steam superheated to 500 deg. fahr. at the superheater. Each division consisted of five series run at different gage pressures at constant speed, and two series run at different speeds with constant pressure. The steam pressures used were 57.5, 76.5, 95, 113, and 132 lb. gage with the engine running at 120 r.p.m. The other speeds employed were 90 r.p.m. and 150 r.p.m. at the gage pressure of 113 lb. Each division, therefore, gave the effect of the use of five steam pressures at constant speed, and three speeds at constant pressure.

19 The governor change-speed device was always set to give the desired speed with the engine running at no-load. As the load was increased, the speed decreased through the action of the governor in about the same proportion for all initial speeds. Whenever speed is mentioned, the no-load speed is the one referred to, the exact speed for any one test being given in the general log.

20 The general log of the 29 tests run with saturated steam is given in Table 1. Table 2 contains the results of the 31 tests run with superheated steam. Table 3 gives the averages of similar series, called groups, run at the same pressure and speed, with both saturated and superheated steam. For any one group of the two series of tests, as has already been pointed out, the only variables are the length of cut-off and the value of X_e.

TABLE 1 GENERAL LOG. TESTS WITH SATURATED STEAM

Series	Number of Test	Pressure Lb. Per Sq. Inch				Revolutions per Hour	Weight of Condensate per Hour, lb.	Weight of Condensate per Revolution, lb.	Lb. of Steam per Revolution shown by Indicator at Cut-Off	Lb. of Compression saved per Revolution by Indicator	Lb. of Steam Mixture Actually present per Revolution	Quality of Steam Mixture at Cut-Off, Parts of Unity	Absolute Pressure at Cut-Off, Lb.	Average Value of n from Expansion Curves	Cut-Off, Per Cent of Stroke	Average Absolute Pressure at Cut-Off for Series, Lb
		At Throttle by Gage	Barometer	Absolute Pressure at Throttle	Average Absolute Pressure of Series at Throttle											
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
I	2	57.8	14.3	72.1	....	6428	1054	0.1640	0.0929	0.0248	0.1888	0.492	60.3	0.926	14.6	....
	3	57.3	14.2	71.5	71.3	6100	1104.5	0.1811	0.1037	0.0221	0.2032	0.510	61.3	0.882	16.5	60.6
	4	55.8	14.5	70.3	....	6302	2010	0.3189	0.2099	0.0276	0.3465	0.606	60.2	0.961	41.2	....
II	6	77.0	14.3	91.3	....	6846	1010	0.1475	0.0846	0.0235	0.1710	0.495	76.8	0.927	8.1	....
	7	76.7	14.2	90.9	90.5	6440	1384	0.2149	0.1281	0.0230	0.2379	0.539	80.2	0.895	15.7	77.9
	8	75.6	14.5	90.1	....	6640	2104	0.3168	0.2089	0.0253	0.3421	0.610	77.4	0.949	30.9	....
	9	75.0	14.5	89.5	....	6282	2559	0.4073	0.2727	0.0260	0.4333	0.629	77.4	0.950	42.6	....
IV	15	95.8	14.3	110.1	....	6940	1048	0.1510	0.0903	0.0236	0.1746	0.517	95.6	0.910	6.1	....
	16	96.0	14.2	110.2	....	6510	1530	0.2350	0.1462	0.0226	0.2756	0.531	95.6	0.887	14.6	....
	17	94.9	14.2	109.1	109.1	6432	1584	0.2462	0.1533	0.0233	0.2695	0.569	96.2	0.914	15.5	95.3
	18	94.5	14.4	108.9	....	6750	2208	0.3404	0.2145	0.0246	0.3650	0.588	92.5	0.996	25.4	....
	19	93.6	14.4	108.0	....	6410	2758	0.4302	0.2952	0.0263	0.4565	0.647	95.6	1.002	37.0	....
	20	94.0	14.4	108.4	....	5958	3008	0.5048	0.3484	0.0248	0.5296	0.658	96.1	0.963	44.3	....
V	22	114.6	14.3	128.9	....	7046	1030	0.1461	0.0907	0.0239	0.1700	0.533	105.5	0.835	4.8	....
	23	113.7	14.2	127.9	127.7	6688	1566	0.2341	0.1445	0.0240	0.2581	0.560	110.0	0.944	11.3	109.4
	24	113.6	14.2	127.8	....	6466	1738	0.2687	0.1776	0.0227	0.2914	0.610	109.1	0.940	15.7	....
	25	112.0	14.2	126.2	....	6736	2664	0.3954	0.2708	0.0253	0.4207	0.644	113.0	1.004	27.3	....
VI	27	131.9	14.3	146.2	....	7086	986	0.1391	0.0941	0.0240	0.1631	0.576	126.7	0.946	3.7	....
	28	133.2	14.2	147.4	....	6862	1466	0.2136	0.1426	0.0231	0.2367	0.602	130.8	0.950	8.3	....
	29	132.0	14.2	146.2	146.2	6306	2000	0.3172	0.2050	0.0220	0.3392	0.604	130.0	1.006	16.4	129.4
	30	130.8	14.2	145.0	....	6840	2796	0.4087	0.2760	0.0256	0.4343	0.635	128.9	1.008	24.9	....
	31	132.0	14.4	146.4	....	6138	3276	0.5337	0.3762	0.0269	0.5606	0.671	130.6	1.010	35.3	....
VII	32	113.9	14.4	128.3	....	5258	1025	0.1949	0.0909	0.0251	0.2200	0.413	112.1	0.854	4.9	....
	33	113.9	14.4	128.3	128.4	5102	1585	0.3106	0.1641	0.0242	0.3348	0.490	113.2	0.932	12.8	113.7
	34	114.1	14.4	128.5	....	4940	2090	0.4230	0.2402	0.0246	0.4476	0.537	115.7	0.930	23.2	....
VIII	36	113.3	14.4	127.7	....	8772	1616	0.1842	0.1136	0.0260	0.2102	0.540	108.4	0.927	7.7	....
	37	114.2	14.4	128.6	127.9	8518	2554	0.2998	0.1990	0.0254	0.3252	0.612	110.3	0.964	19.7	110.3
	38	112.8	14.4	127.2	....	8182.5	3520	0.4302	0.3080	0.0256	0.4558	0.676	110.1	1.045	34.4	....
	39	113.5	14.4	127.9	....	7625	4142.5	0.5433	0.3968	0.0259	0.5692	0.697	112.2	1.016	46.0	....

TABLE 2 GENERAL LOG. TESTS WITH SUPERHEATED STEAM

Series	Number of Test	Pressure Lb. Per Sq. Inch: At Throttle by Gage	Pressure Lb. Per Sq. Inch: Barometer	Pressure Lb. Per Sq. Inch: Absolute Pressure at Throttle	Pressure Lb. Per Sq. Inch: Average Absolute Pressure of Series at Throttle	Revolutions per Hour	Weight of Condensate per Hour, Lb.	Weight of Condensate per Revolution, Lb.	Lb. of Steam per Revolution shown by Indicator at Cut-off	Lb. of Compression saved per Revolution by Indicator	Lb. of Steam Mixture Actually present per Revolution	Quality of Steam Mixture at Cut-Off, Parts of Unity	Absolute Pressure at Cut-Off, Lb.	Average Value of n from Expansion Curves	Cut-Off, Per Cent of Stroke	Average Absolute Pressure at Cut-Off for Series, Lb.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
IX	40	59.5	14.5	74.0	....	7040	815	0.1158	0.0838	0.0232	0.1390	0.602	61.2	0.904	11.7	....
	41	60.5	14.5	75.0	73.4	6856	1058	0.1543	0.1235	0.0239	0.1782	0.693	61.6	1.053	21.6	61.1
	42	57.7	14.5	72.2	....	6522	1288	0.1975	0.1738	0.0228	0.2203	0.788	60.6	1.114	32.0	....
	43	57.8	14.5	72.3	....	6262	1396	0.2229	0.2110	0.0227	0.2456	0.859	60.8	1.144	41.3	....
X	44	76.1	14.5	90.6	....	6655	1201.7	0.1806	0.1106	0.0246	0.2052	0.539	79.2	0.951	12.4	....
	45	77.4	14.5	91.9	91.0	6550	1294	0.1976	0.1342	0.0250	0.2226	0.603	79.2	0.957	16.5	78.5
	46	76.3	14.5	90.8	....	6290	1455	0.2313	0.1910	0.0230	0.2543	0.751	77.4	1.105	27.6	....
	47	76.0	14.5	90.5	....	5908	1822	0.3084	0.2720	0.0226	0.3310	0.822	78.0	1.112	41.5	....
XI	48	94.3	14.5	108.8	....	7112	822	0.1156	0.0877	0.0228	0.1384	0.634	92.8	0.924	6.1	....
	49	93.0	14.5	107.5	....	6936	1270	0.1831	0.1311	0.0234	0.2065	0.635	90.7	0.982	13.3	....
	50	95.5	14.5	110.0	109.0	6682	1660	0.2484	0.2040	0.0226	0.2710	0.753	97.1	1.040	22.3	94.1
	51	94.7	14.4	109.1	....	6500	1780	0.2738	0.2479	0.0220	0.2958	0.838	96.4	1.116	28.8	....
	52	95.1	14.4	109.5	....	6240	2072	0.3321	0.2821	0.0226	0.3547	0.796	93.6	1.001	39.2	....
XII	53	115.2	14.4	129.6	....	7100	914	0.1287	0.0972	0.0224	0.1511	0.642	111.5	0.991	5.1	....
	54	114.3	14.4	128.7	....	6932	1380	0.1991	0.1561	0.0225	0.2216	0.705	109.0	0.990	13.0	....
	55	114.1	14.4	128.5	128.4	6662	1906	0.2861	0.2419	0.0226	0.3087	0.783	111.7	1.032	22.9	111.0
	56	114.1	14.3	128.4	....	6326	2315	0.3660	0.3230	0.0219	0.3879	0.833	111.5	1.140	35.1	....
	57	112.3	14.5	126.8	....	5940	2350	0.3956	0.3709	0.0228	0.4184	0.886	111.5	1.209	40.3	....
XIV	61	132.5	14.3	146.8	....	7136	922	0.1292	0.0935	0.0228	0.1520	0.615	127.7	0.939	3.2	....
	62	128.0	14.3	142.3	....	7010	1384	0.1974	0.1553	0.0229	0.2203	0.715	123.6	0.969	11.0	....
	63	131.3	14.3	145.6	145.4	6810	1789	0.2627	0.2331	0.0228	0.2855	0.817	131.0	1.074	18.1	128.5
	64	131.0	14.3	145.3	....	6602	2171	0.3288	0.3113	0.0222	0.3510	0.887	130.5	1.190	26.9	....
	65	132.6	14.5	147.1	....	6088	2640	0.4336	0.3988	0.0230	0.4566	0.873	129.5	1.165	37.1	....
XV	66	114.8	14.5	129.3	....	5322	808	0.1518	0.0950	0.0220	0.1738	0.546	112.3	0.948	4.6	....
	67	114.2	14.5	128.7	128.0	5212	1218	0.2337	0.1540	0.0222	0.2559	0.602	111.1	0.941	12.4	111.7
	68	112.3	14.5	126.8	....	5018	1574	0.3137	0.2356	0.0226	0.3363	0.700	110.7	1.013	23.1	....
	69	112.7	14.5	127.2	....	4817	1809	0.3755	0.3078	0.0217	0.3972	0.774	112.5	1.148	32.0	....
XVI	71	112.8	14.5	127.3	....	8700	1512	0.1738	0.1447	0.0249	0.1987	0.729	108.0	1.093	12.3	....
	72	111.8	14.5	126.3	126.9	8438	2018	0.2392	0.2080	0.0257	0.2649	0.786	109.0	1.091	20.5	108.3
	73	112.2	14.5	126.7	....	8084	2458	0.3041	0.2970	0.0236	0.3277	0.907	107.2	1.213	32.4	....
	74	112.6	14.5	127.1	....	7494	3110	0.4150	0.4135	0.0237	0.4387	0.943	108.8	1.234	46.9	....

TABLE 3 LOG OF GROUP AVERAGES

Group	Series	Number of Tests Inclusive	Pressure, Lb. per Sq. In.			Absolute Pressure from Indicator Diagram, Lb. per Sq. In.					Specific Volume of Steam at Pressure Adopted	Number of Tests in Series	Number of Tests in Group	Number of Diagrams Analyzed	Average Quality of Steam Mixture at Cut-Off from Tests in Group, Parts of Unity	Average Value of n from Tests in Group
			Average Absolute Pressure at Throttle by Gage	Average Absolute Pressure of Group	Difference from Group Average of Next Lower Pressure	Average at Cut-Off	Average of Group	Difference from Group Average of Next Lower Pressure	Drop in Pressure to Cut-Off	Pressure Adopted to find Quality of Steam Mixture at Cut-Off						
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
A	I	2-4	71 3	72 4		60 6	60 9		11 5	61 0	7 06	3	7	14	0 6569	0 998
	IX	40-43	73 4			61 1						4				
B	II	6-9	90 5	93 8	18 4	77 9	78 2	17 3	12 6	78 0	5 60	4	8	16	0 6235	0 981
	X	44-47	91 0			78 5						4				
C	IV	15-20	109 1	109 1	15 3	95 3	94 7	16 5	13 4	95 0	4 65	6	11	22	0 6515	0 976
	XI	48-52	108			94 1						5				
D	V	22-25	127 7	127 9	18	109 4	110 7	16 0	17 2	111 0	4 012	5	9	18	0 6885	1 010
E	VII	32-34	128 3			110 7						5				
D	VIII	36-39	127 9			110 3						5		14	0 5803	0 997
E	XII	53-57	126 4			110 0						5	7	16	0 7393	1 073
F	XV	66-68	126 0			111 3						4				
G	XVI	71-74	126 5			108 3						4				
	VI	27-31	146 2	145 8	17 9	129 3	129 0	18 3	16 8	129 0	3 378	5	10	20	0 6995	1 026
	XIV	61-65	145 4			128 5						5				

21 *Values of n obtained under Different Conditions from the Same Engine Cylinder.* All the simultaneous values of X_c and n obtained from the 60 tests were plotted in Fig. 1. A study of this figure shows beyond question that as X_c increases in value, n increases also. The values all lie in a region which has a definite trend towards higher simultaneous values of X_c and n. Observing the general trend, it is seen that there is no exception to this general relation. No value of n below 1.00, for instance, is found for values of X_c above 0.80, and no value of n above 1.10 is found for values of X_c below 0.72.

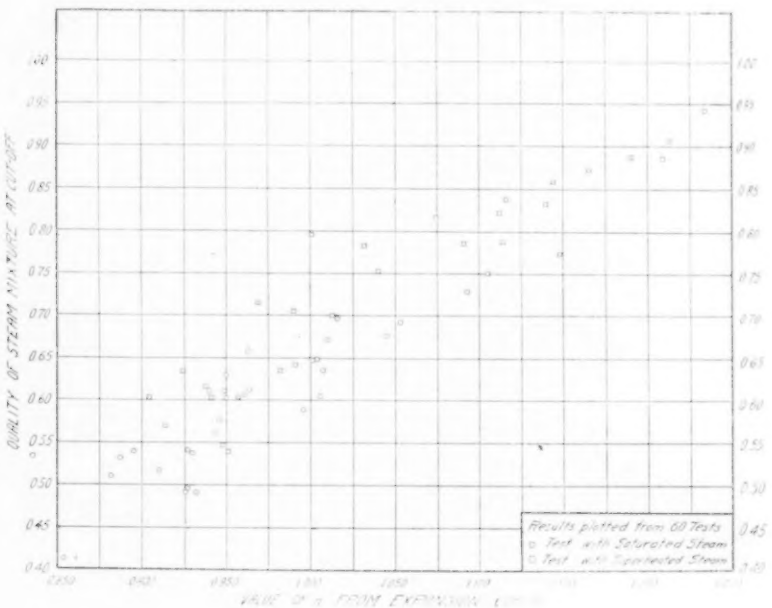

FIG. 1 GENERAL RELATIONS BETWEEN QUALITY AT CUT-OFF AND THE VALUE OF n FOR VARIOUS PRESSURES AND SPEEDS

22 It is also apparent that the points with long cut-off, obtained from saturated steam for given values of X_c and n, are in the same region occupied by the points with short cut-off obtained from superheated steam for the same given values of X_c and n. Examining the region of $n = 0.90 - 1.00$ and of $X_c = 0.50 - 0.70$, we see that the points obtained with long cut-off with saturated steam, and with short cut-off with superheated steam, lie indiscriminately together. This shows conclusively, in a general way, that the value of n is practically independent of the length of cut-off, even though this

length may vary from 5 per cent to 45 per cent, and that n depends solely on the value of X_e, the only other variable.

23 The points shown in Fig. 1 occupy a relatively wide region until they are separated into the various groups of similar pressures and speeds.

24 The points for each group were plotted separately, and separate curves were determined for each condition. From preliminary plotting, the relations between n and X_e were found to be expressed

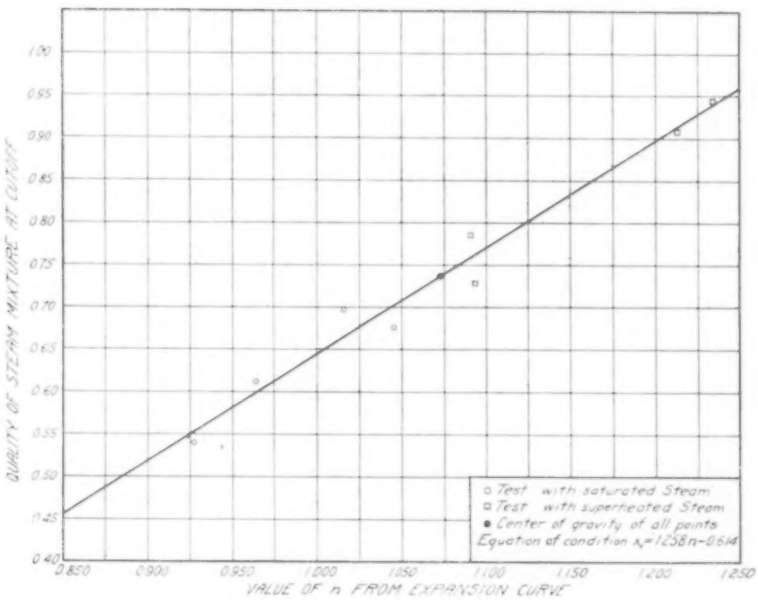

FIG. 2 RELATION OF QUALITY AND THE VALUE OF n FOR TESTS RUN AT 111 LB. ABSOLUTE PRESSURE OF CUT-OFF AND 150 R.P.M.

closer by straight lines than by any other family of curves. The method used to draw these lines will be given for group F, which includes series VIII, with four tests, and series XVI with four tests. This group is shown in Fig. 2. All points were given equal weight. The average of all the coördinates, or the "center of gravity" was found and the conditions imposed that the line pass through this center as an axis, and that the slope be determined by the position of the points. The points in group F were divided into four logical pairs or groupings, and the center of gravity found for each grouping. The line was then drawn as shown. Where points were located

so that a logical grouping was in doubt, various groupings were made, and each given weight in determining the slope.

25 The equation of the curve selected is

$$X_c = 1.258\, n - 0.614$$

The average deviation of the points from this line is 2.6 per cent (measured from the zero of X_c) and the maximum deviation is 4.6 per cent. This average deviation, 2.6 per cent, is smaller than that for most of the groups. The straight line, in most cases, represents the points found as close as any other curve that could be employed, and has the merit of simplifying greatly the subsequent use made of the relations for the different groups.

26 The values of n for group F were also plotted to the various accompanying cut-off positions at which each test was run. This is shown in Fig. 3. The points in Fig. 3 show that for each initial condition of steam the value of n increased as the cut-off was lengthened, and that with any given cut-off different values of n were obtained according as saturated or superheated steam was used. Thus at a cut-off of 15 per cent, the value of n obtained is 0.950 with saturated steam, and 1.084 with superheated steam. The only variable present in these two cases is X_c, the value of which is higher with superheated steam than with saturated steam. A definite relation between n and the length of cut-off occurs only when a given cut-off is accompanied by the same value of X_c, that is, the value of n bears a direct relation to X_c but not to cut-off. Fig. 3 taken in conjunction with Fig. 2 proves that the value of n depends directly only upon the value of X_c, and that the relation of n and X_c is practically independent of the length of cut-off within the limits of the tests.

27 *Effect of Varying the Steam Pressure at Constant Speed.* The relations of X_c and n were determined separately for all groups by the method outlined for Fig. 2. The lines for the five pressures used, comprising the results of groups A, B, C, D and G, were replotted as shown in Fig. 4. This figure also contains other curves that are discussed in Paragraph 38. The lines shown give the relations of X_c and n for various absolute pressures at cut-off, all obtained with a speed of 120 r.p.m.

28 These curves were then examined to find the effect of varying the absolute pressure at cut-off (designated as p) on the relations of X_c and n. In Fig. 4, the constant pressure curves were intercepted at constant values of n, and the coördinates of X_c and p for the points of intersection plotted in Fig. 5. This process was repeated at intervals of 0.05 for the values of n from 0.850 to 1.250. The

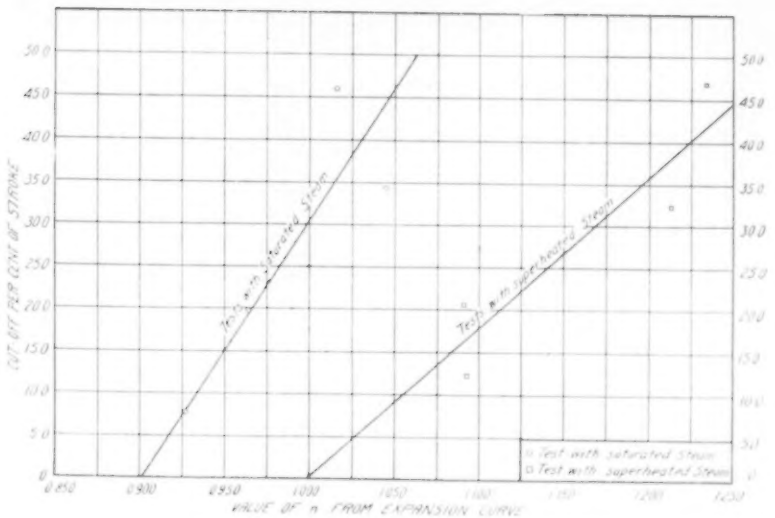
CUT-OFF PER CENT OF STROKE
VALUE OF n FROM EXPANSION CURVE
Test with saturated Steam
Test with superheated Steam
Test with saturated Steam
Test with superheated Steam

FIG. 3 RELATIONS OF PER CENT OF CUT-OFF AND THE VALUE OF n FOR TESTS RUN AT 111 LB. ABSOLUTE PRESSURE AT CUT-OFF AND 150 R.P.M.

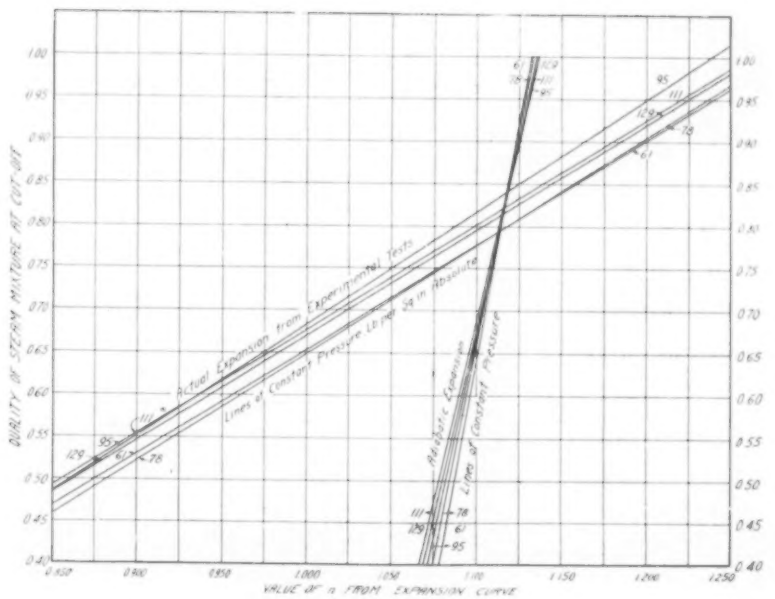
QUALITY OF STEAM MIXTURE AT CUT-OFF
VALUE OF n FROM EXPANSION CURVE
Actual Expansion from Experimental Tests
Lines of Constant Pressure Lbs. per sq in Absolute
Adiabatic Expansion
Lines of Constant Pressure

FIG. 4 RELATIONS AT CUT-OFF, BETWEEN QUALITY AND THE VALUE OF n FOR VARIOUS PRESSURES AT CONSTANT SPEED

resulting points did not give smooth curves, but, after considerable study and trial, curves were adopted as shown.

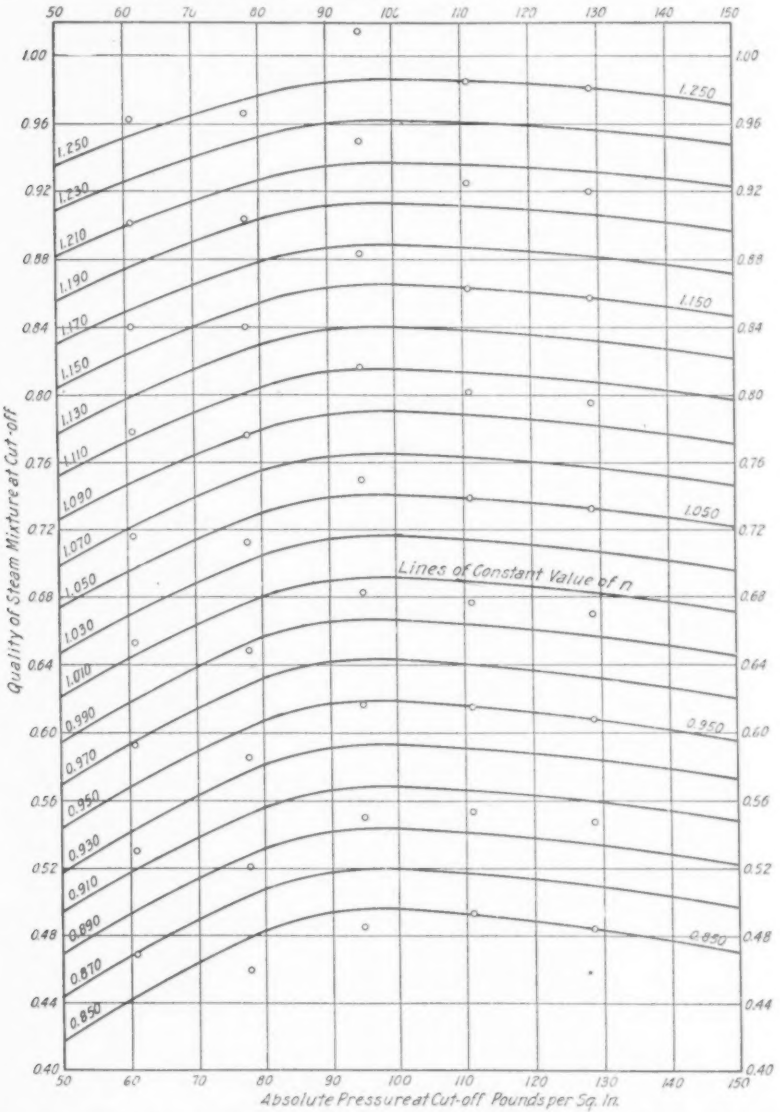

FIG. 5 THE RELATION AT CUT-OFF, BETWEEN QUALITY AND PRESSURE FOR CONSTANT VALUES OF n FROM EXPANSION CURVE

F 29 This procedure gave a series of relations between X_c and p for constant values of n. Since, however, the independent variables

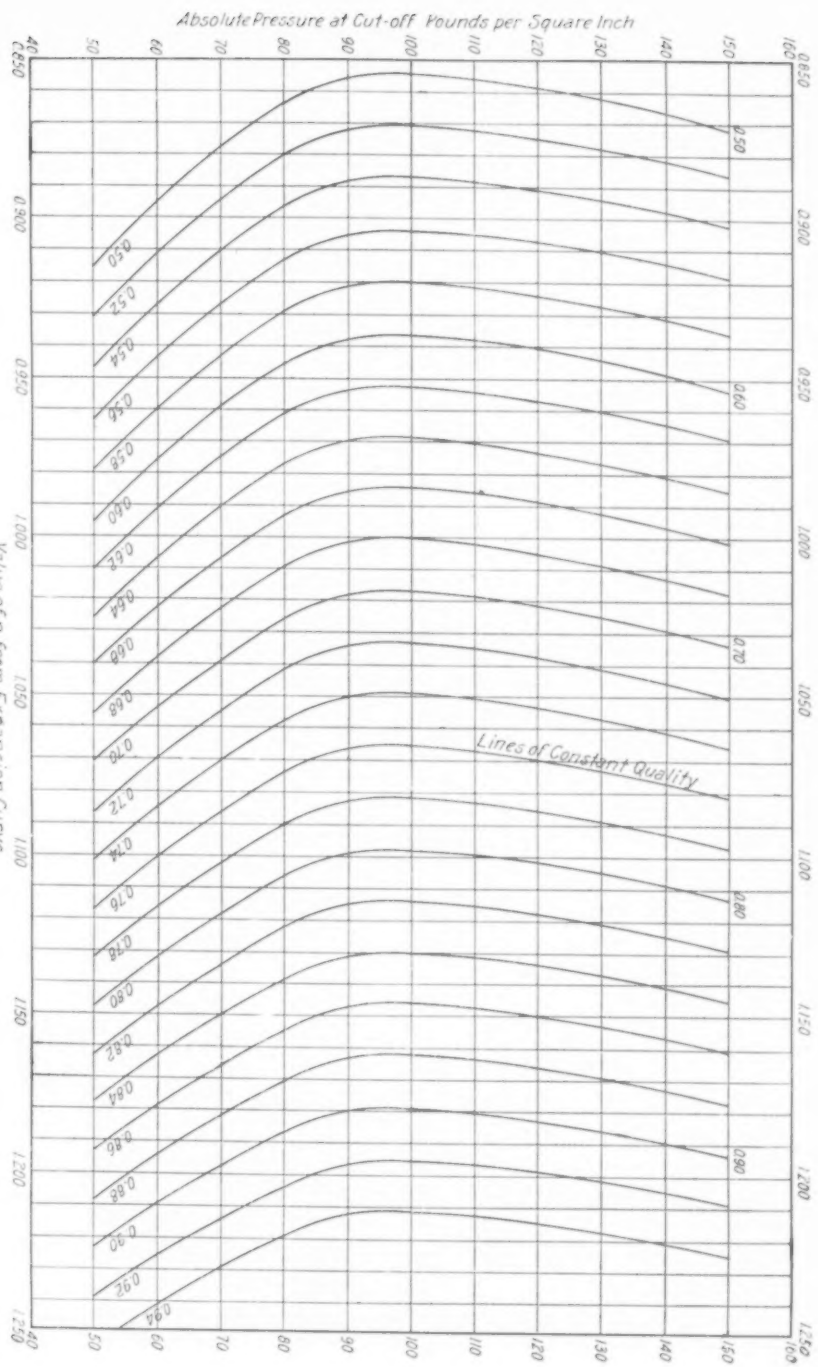

FIG. 6 THE RELATION BETWEEN PRESSURE AT CUT-OFF AND VALUES OF n FOR CONSTANT QUALITY AT CUT-OFF

in any actual curve under examination are n and p, the coördinates of the curves of Fig. 5 were changed so as to show the relations of n and p at constant values of X_c. These are shown in Fig. 6.

30 The effect of a change of pressure on the relations of X_c and n is not great between the limits of 75 to 150 lb. An approximate equation has been worked out, therefore, which represents the relations of X_c and n at an average pressure between the limits mentioned. The equation corresponds to the relations at 129 lb. and is of the form

$$X_c = 1.245\ n - 0.576$$

31 *Effect of Varying the Speed at Constant Pressure.* Groups *D*, *E*, and *F* were run at speeds of 120, 90, and 150 r.p.m., respectively, with the average cut-off pressure on the diagrams constant at 111 lb. absolute.

32 The regulation of the governor was very poor, there being about a 10 per cent drop in speed from no-load to full load. For this reason the relations of X_c and n with various speeds at constant pressure were affected by considerable variation of the speed itself for each group.

33 The relations for each group were found as already described, and the curves plotted in Fig. 7. The relation of speed (designated as s) and X_c for constant values of n was derived from Fig. 7, and is given in Fig. 8.

34 The apparent relations of X_c, s, and n, obtained by drawing a smooth curve through the three points obtained for each value of n, is not satisfactory owing to insufficient data and the change of the speed itself in the three groups due to poor regulation.

35 The drop in the speed, for one group, does not seriously affect, however, the relations of X_c and n for the various pressures at constant speed.

36 *Relation of the Value of n to the Quality of the Steam Mixture at Cut-Off.* From the evidence obtained from these tests, it may be stated positively that for any one engine running at a given pressure and speed, there is a definite relation between X_c and n which is practically independent of the cut-off position within the limits examined. This relation is apparently a linear one. It may also be stated that the relation of X_c and n is dependent, to some extent, on the absolute pressure at cut-off, and on the speed of the engine.

37 It remained to compare the relations of X_c and n for the engine tested with the relations for other engines. This comparison is made in Table 4.

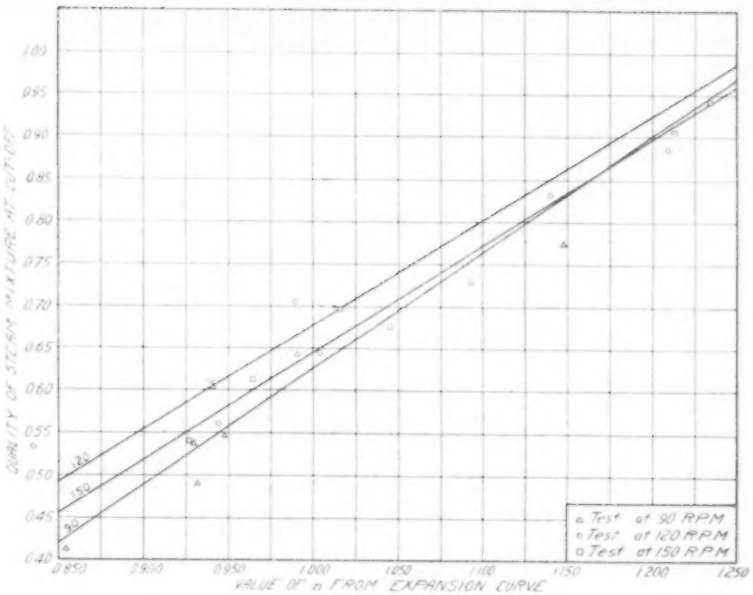

FIG. 7 RELATION OF QUALITY AT CUT-OFF AND THE VALUES OF n FOR VARIOUS SPEED OF THE CONSTANT CUT-OFF PRESSURE OF 111 LB.

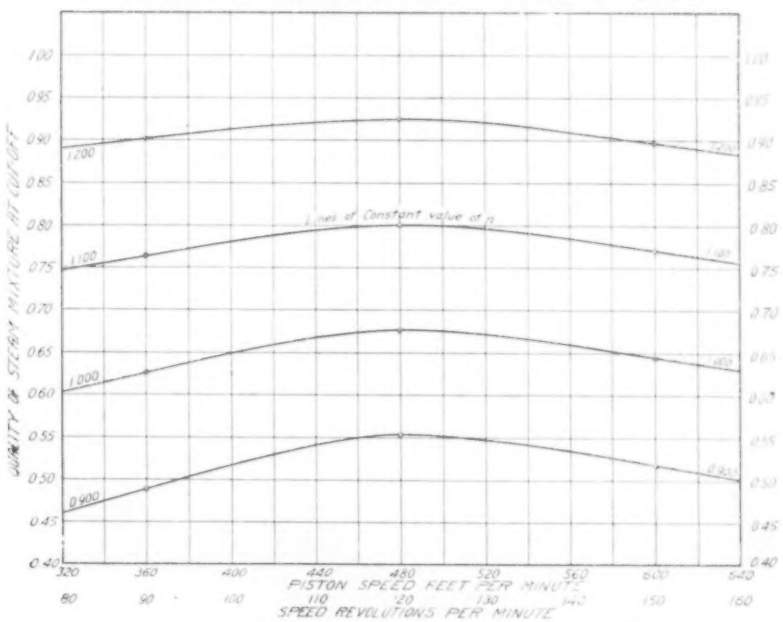

FIG. 8 RELATIONS, AT CUT-OFF, BETWEEN QUALITY AND SPEED FOR CONSTANT VALUES OF n AT CONSTANT PRESSURE

TABLE 4 APPROXIMATION OF THE ACTUAL STEAM CONSUMPTION FROM INDICATOR DIAGRAMS TAKEN FROM NON-JACKETED STEAM ENGINES

Number of Analysis	Class of Engine	Cylinder Dimensions, In.	Piston Rod Diameter, In.	Clearance, Per Cent of Piston Displacement	Actual Weight of Steam Passing through Cylinder, Lb. per Hour	Total I.H.P. Developed	Actual Steam per I.H.P.-Hr., Lb.	Revolutions per Minute	Revolutions per Hour	Weight of Steam		
										Passing through Cylinder per Revolution, Lb.	Shown by Indicator at Cut-Off, Lb.	Retained in Compression per Revolution by Indicator, Lb.
1	2	3	4	5	6	7	8	9	10	11	12	13
	Simple Corliss, 2-Valve, and 4-Valve, Non-Condensing Engines											
101	Corliss, single cylinder	23 x60	$3\frac{1}{4}$	2.5	8477	305.2	27.77	74.7	4482	1.890	1.666	0.130
102	Corliss, single cylinder	$28\frac{1}{2}$x$59\frac{1}{2}$	4	3.0	13070	506.5	25.80	64.8	3888	3.362	2.822	0.167
103	Corliss, single cylinder	$26\frac{7}{8}$x48	$3\frac{7}{8}$	3.0	6742	232.3	29.03	64.7	3882	1.736	1.402	0.116
104	Double-valve, 2 cylinders	17 x24.2	$2\frac{3}{4}$	2.0	7952	310.1	25.64	152.9	9174	0.8670	0.7290	0.0635
105	Four-valve, single cylinder	$16\frac{5}{8}$x32	$2\frac{3}{4}$	4.0	1879	50.2	37.43	79.8	4788	0.3925	0.2732	0.0607
106*a*	Four-valve, single cylinder	$10\frac{1}{2}$x12	2	5.0	2353	97.9	24.04	263	15780	0.1491	0.1377	0.0242
106*b*	Same engine as 106*a*	.....	...	....	1878	79.2	23.71	258	15480	0.1213	0.0959	0.0160
106*c*	Same engine as 106*a*	.....	...	....	997	41.7	23.90	260	15600	0.0639	0.0537	0.0260
106*d*	Same engine as 106*a*	.....	...	....	693	25.8	26.87	263	15780	0.0439	0.0402	0.0351
107	Corliss, 2 cylinders	16 x42	$2\frac{3}{4}$	2.5	8873	342.43	25.91	84.9	5094	1.742	1.420	0.041
108*a*	Four-valve, single cylinder	18 x30	$2\frac{5}{8}$	5.0	4707	213.2	22.08	165.6	9936	0.4735	0.3890	0.0330
108*b*	Same engine as 108*a* (condensing)	.....	...	....	6050.2	209.1	28.93	164.4	9864	0.6137	0.5657	0.0712
109	Corliss, 2 cylinders (1 condensing)	23 x60	$3\frac{3}{8}$	3.0	13187	615.1	21.44	61	3660	3.602	3.140	0.080
110	Corliss, 1 cylinder (1 end condensing)	34.2x60	$4\frac{7}{8}$	3.0	11613	523.43	22.19	53.3	3198	3.633	2.835	0.093
111	Corliss, 2 cylinders (1 condensing)	28 x60	4	2.5	14295.7	562.1	25.43	60.27	3616.2	3.953	2.860	0.250
	Compound Condensing Engines—H.P. Cylinders											
151	Corliss cross-compound	20 x36x36	3	2.5	6740	433.5	15.00	75.83	4550	1.481	1.244	0.1012
152	Buckeye cross-compound	$18\frac{3}{4}$x36x36	$3\frac{1}{2}$	4.3	8675	639.2	14.20	123.3	7398	1.173	1.023	0.1490
153	Corliss cross-compound	15x$30\frac{1}{8}$x30	$2\frac{3}{4}$	4.0	2007	140.78	14.26	47.98	2879	0.6970	0.5492	0.0218
	Triple-Expansion Condensing Engines—I. P. Cylinders											
201	Corliss (slow-speed pumping engine)	$24\frac{1}{8}$x34x54x36	$4\frac{1}{2}$	2.10	4622	328.05	14.09	27.66	1659.6	2.787	1.432	0.031
	High-Speed Single-Valve, Non-Condensing Engines											
401	Piston valve, single cylinder	16 x16	$2\frac{5}{8}$	13.15	4820	184.6	26.35	253.4	15204	0.3196	0.3060	0.0948
402	Single valve, single cylinder	$14\frac{1}{2}$x13	$2\frac{1}{4}$	10.0	1740	53.26	32.67	246	14760	0.1179	0.1197	0.0613
403	Single valve, single cylinder	$14\frac{1}{2}$x13	$2\frac{1}{4}$	12.0	2018.6	61.7	32.71	248.4	14904	0.1353	0.1314	0.0594
	Simple Locomotive Engines			H C								
451	Two cylinder slide valve	16 x24	$2\frac{3}{4}$	R 7.44 7.98 L 7.34 7.63	7180	252.79	28.40	97.08	5824.8	1.232	1.0164	0.1947

Number of Analysis.	Weight of Steam: Mixture Actually Present per Revolution, Lb.	Quality of Steam Mixture at Cut-Off (X_c), Parts of Unity	Average Value of n from Expansion Curves	Absolute Pressure at Cut-Off, Lb. per Sq. In.	Approximation from Chart: Value of X_c from Chart, Parts of Unity	Approximation from Chart: Per Cent that the Value of X_c from Chart is Higher (+) or Lower (−) than Test Results	Approximation from Chart: Total Weight of Steam Mixture Present per Revolution, Lb.	Approximation from Chart: Weight of Steam: Passing through Cylinder per Revolution, Lb.	Approximation from Chart: Weight of Steam: Passing through Cylinder per Hour, Lb.	Approximation from Chart: Weight of Steam: Per I.H.P-hr., Lb.	Approximation from Chart: Per Cent that Total Weight of Steam per Hour is Higher (+) or Lower (−) than Test Results	Approximation from Equation $X_c = 1.245n - 0.576$: Value of X_c from Equation $X_c = 1.245n - 0.576$	Approximation from Equation $X_c = 1.245n - 0.576$: Per Cent that the Value of X_c from Equation is Higher (+) or Lower (−) than Test Results	Approximation from Equation $X_c = 1.245n - 0.576$: Total Weight of Steam Mixture present per Revolution, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Passing through Cylinder per Revolution, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Passing through Cylinder per Hour, Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Weight of Steam: Per I.H.P-Hr., Lb.	Approximation from Equation $X_c = 1.245n - 0.576$: Per Cent that Total Weight of Steam per Hour is Higher (+) or Lower (−) than Test Results	Method Giving Closer Approximation to Test Results, Chart (*C*), Equation (*E*)
1	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Simple Corliss, 2-Valve, and 4-Valve, Non-Condensing Engines																			
101	2.020	0.825	1.098	65	0.767	−7.0	2.172	2.042	9160	30.00	+8.1	0.790	−4.2	2.110	1.980	8880	29.18	+4.8	*E*
102	3.529	0.801	1.108	75	0.795	−0.7	3.551	3.384	13150	25.96	+0.7	0.804	+0.4	3.512	3.345	13000	25.57	−0.5	*E*
103	1.852	0.757	1.051	75	0.732	−3.3	1.915	1.799	6985	30.06	+3.6	0.733	−3.2	1.913	1.797	6980	30.02	+3.5	*E*
104	0.9305	0.784	1.131	70	0.816	+4.1	0.8935	0.8300	7615	24.53	−4.2	0.833	+6.3	0.8752	0.8117	7445	24.00	−6.4	*C*
105	0.4532	0.603	0.964	60	0.586	−2.8	0.4665	0.4058	1942	38.69	+3.3	0.624	+3.5	0.4381	0.3774	1807	36.00	−3.8	*C*
106*a*	0.1733	0.794	1.076	120	0.769	−3.1	0.1791	0.1549	2440	24.92	+3.7	0.764	−3.8	0.1802	0.1560	2461	25.15	+4.6	*C*
106*b*	0.1373	0.698	1.034	115	0.717	+2.7	0.1337	0.1177	1822	23.01	−3.0	0.711	+1.9	0.1349	0.1189	1840	23.22	−2.0	*E*
106*c*	0.0899	0.598	0.931	125	0.587	−1.8	0.0915	0.0655	1021	24.50	+2.4	0.583	−2.5	0.0921	0.0661	1031	24.72	+3.4	*C*
106*d*	0.0790	0.509	0.868	120	0.511	+0.4	0.0787	0.0436	688	26.68	−0.7	0.504	−1.0	0.0798	0.0447	705	27.32	+1.2	*C*
107	1.783	0.797	1.077	95	0.774	−2.9	1.835	1.794	9140	26.69	+3.0	0.765	−4.0	1.856	1.815	9245	27.00	+4.2	*C*
108*a*	0.5065	0.768	1.095	63	0.759	−1.2	0.5125	0.4795	4762	22.32	+1.2	0.788	+2.6	0.4937	0.4607	4577	21.45	−2.8	*C*
108*b*	0.6849	0.826	1.116	63	0.785	+5.0	0.7202	0.6490	6402	30.60	+5.8	0.814	−1.5	0.6947	0.6235	6150	29.40	+1.7	*E*
109	3.682	0.853	1.149	80	0.853	±0.0	3.682	3.602	13187	21.44	±0.0	0.854	+0.1	3.677	3.597	13160	21.40	−0.2	*C*
110	3.726	0.761	1.071	74	0.749	−1.6	3.785	3.692	11810	22.57	+1.7	0.758	−0.4	3.740	3.647	11655	22.28	+0.4	*E*
111	4.203	0.680	1.048	68	0.709	+4.3	4.033	3.783	13685	24.33	−4.3	0.729	+7.2	3.922	3.672	13280	23.62	−7.1	*C*
Compound Condensing Engines—H.P. Cylinders																			
151	1.582	0.786	1.079	100	0.777	−1.1	1.600	1.4988	6820	15.73	+1.2	0.767	−2.4	1.621	1.5198	6915	15.95	+2.6	*C*
152	1.322	0.774	1.060	120	0.749	−3.2	1.366	1.2170	9005	14.09	+3.8	0.744	−3.9	1.375	1.226	9070	14.19	+4.6	*C*
153	0.7188	0.764	1.047	130	0.728	−4.7	0.7542	0.7324	2109	14.97	+5.1	0.728	−4.7	0.7542	0.7324	2109	14.97	+5.1	*C* & *E*
Triple Expansion Condensing Engines—I.P. Cylinders																			
201	2.818	0.508	0.914	38.5	0.470	−7.5	3.048	3.017	5003	15.25	+8.2	0.562	+10.6	2.550	2.519	4180	12.74	−9.6	*C*
High-Speed Single Valve Non-Condensing Engines																			
401	0.4144	0.739	1.027	110	0.710	−3.9	0.4310	0.3362	5115	27.70	+6.1	0.703	−4.9	0.4352	0.3404	5178	28.02	+7.4	*C*
402	0.1792	0.668	0.970	85	0.637	−4.6	0.1879	0.1266	1870	35.10	+7.5	0.631	−5.5	0.1898	0.1285	1897	35.61	+9.0	*C*
403	0.1947	0.675	0.985	74	0.641	−5.0	0.2050	0.1456	2170	35.19	+7.5	0.650	−3.7	0.2021	0.1427	2128	34.48	+5.4	*E*
Simple Locomotive Engines																			
451	1.4267	0.711	1.003	100	0.683	−3.9	1.488	1.2933	7540	29.83	+5.0	0.673	−5.3	1.5110	1.3163	7675	30.38	+6.9	*C*

38 An investigation [1] of the value of k for adiabatic expansion shows that there is a relation between the initial quality X_1 and the value of k which, like the experimentally determined relation, is also a linear one. The adiabatic relations of X and k are plotted in Fig. 4 for the pressures used in the tests.

THE METHOD OF APPROXIMATING THE ACTUAL STEAM CONSUMPTION FROM INDICATOR DIAGRAMS

39 When indicator diagrams were first obtained from steam engines, loss from initial condensation was not suspected and it was believed that the steam consumption could be determined from the diagram at the point of cut-off. After the classic tests of Clark and Isherwood, however, the existence and amount of this initial condensation was revealed; but the great difference in the proportion that the initial condensation bears to the total weight of mixture present, either at cut-off or during the expansion, as found in different types and sizes of engines, has prevented any reliable determinations of the actual steam consumption by this method. The steam consumption computed from the diagram, when using saturated steam, is generally from 15 per cent to 50 per cent below the actual consumption.

40 The devising of an accurate method of measuring the actual weight of steam consumed from the diagram has therefore been regarded as impossible. Thurston [2] states that "the steam or water consumption of an engine cannot be exactly ascertained by the use of the indicator" for the reasons mentioned. Most of the other writers on the subject have expressed smilar views.

41 Methods of computing the weight or proportion of the initial condensation from the dimensions, type, and speed of the engine considered, as shown by the results of large numbers of tests, have been devised by many.[3] The results obtained from these methods have not been uniform, and do not agree closely enough with the test results to be used with confidence. Moreover, none of these methods is applicable when superheated steam is used. R. C. H.

[1] Principles of Thermodynamics, G. A. Goodenough, p. 191.

[2] Engine and Boiler Trials, R. H. Thurston, p. 237.

[3] A Manual of the Steam Engine, R. H. Thurston, p. 517.
Engineer (London), Escher, 1882.
Relative Proportions of the Steam Engine, L. S. Marks, p. 206.
Proc. Inst. Mech. Engrs., October 1889.
Engineering, Bodmer, March 4, 1892, p. 299.
The Steam Engine, Cotterill, p. 339.

Heck [1] states that the steam consumption computed by the use of his formula should ordinarily show not more than 10 per cent difference from the test results.

42 *The Phenomena Occurring in the Cylinder.* The greatest source of loss in the cylinder is due primarily from practical considerations to the use of a metallic structure. The skin surface of this metal, which is a fairly good conductor of heat, must be heated once every cycle from the temperature acquired from contact with the exhaust steam, nearly up to the temperature of the admission steam, this heating being accomplished by the condensation of some of the incoming steam.

43 The amount of this condensation, measured as the proportion of the mixture present, varies with the size, valve design, relative roughness of the interior surface, temperature range, length of cut-off, speed, location of ports and port passages, quality of the steam supplied, and the jacketing and lagging. It can easily be seen, from the number and relative magnitude of these variables, that the computation of the weight of condensation, by means of a formula which will take these variables into account, can never be an accurate operation.

44 After many examinations into the cases of different types and sizes of engines, with non-jacketed cylinders in good order, and with pressure limits similar to those used with the tests, it has been found that while the initial condensation is subject to the action of ten or more variables, yet the value of n resulting from a given value of X_e is almost always substantially the same. A few of the applications showing this point will be found in Table 4. Here the cylinder sizes vary from $10\frac{1}{2}$ in. by 12 in. to 34.2 in. by 60 in., the speeds from 48 to 263 r.p.m., and the types include slow-speed Corliss, high-speed, and locomotive engines. The possibility of calculating accurately the weight of condensation in these different cases may be easily imagined.

45 The phenomena caused by the presence of the cylinder walls in the class of engines discussed have been found to be divided into two natural classes: those occurring before cut-off, and those occurring after cut-off.

46 The phenomena occurring before cut-off are controlled by the action of the ten or more variables already mentioned, and therefore are subject to all the variation that may occur in any individual case to be examined. For this reason any method of computing the

[1] The Steam Engine, R. C. H. Heck, p. 119.

condensation accurately from the physical facts surrounding the case is open to objection. This method also cannot allow for the use of superheated steam, an increasingly important condition.

47 The phenomena occurring after cut-off are practically independent of all variables except X_e, and the initial pressure and speed. Of these variables, only the value of X_e and the initial pressure have proved to be of material importance in the applications made thus far.

48 This may be summed up by stating that the value of X_e, in any particular case, is subject to the action of many important variables, but that the relation of X_e and n is practically independent of these variables within the limits examined in this investigation.

49 *The Phenomena of Condensation and Re-Evaporation during Expansion.* When adiabatic expansion of initially dry saturated steam takes place, a part of the steam is condensed as the pressure is lowered, the condensed steam giving up its latent heat which is converted into work. When superheated steam is expanded adiabatically, the steam loses its superheat until saturation is reached, after which condensation takes place as in the case of initially dry steam.

50 When, however, the steam is initially composed of a large proportion of water, both being at the same temperature, adiabatic expansion may take place without additional condensation and may even be accompanied by re-evaporation. This fact is due to the large amount of heat contained in the water, a part of which flashes into steam as the pressure is lowered, thus supplying and neutralizing the loss of steam volume by condensation which takes place with steam initially dry. Adiabatic expansion is accompanied by condensation when the initial quality is above the value 0.50 at an initial pressure of 240 lb. per sq. in. absolute, but below the value of 0.50 it is accompanied by re-evaporation. An examination of the temperature entropy diagram for steam will show the values of the initial quality which form the line of demarcation of condensation and re-evaporation during the adiabatic change of state.

51 In the actual engine using saturated steam, as has already been pointed out, some of the incoming steam is condensed in warming up the skin surface of the cylinder walls to approximately the temperature of the incoming steam. When the admission of steam is cut off and expansion commences, the condensation, due to the presence of the cylinder walls, continues in general until, at some point during expansion, the water on the cylinder walls begins to re-

evaporate at such a rate that the weight of steam present at the end of expansion is greater than that which was present at cut-off.

52 To show the effects and extreme values of condensation and re-evaporation during expansion, there have been prepared in Table 5 two examples, using the average results of the tests run.

TABLE 5 CONDENSATION AND RE-EVAPORATION OF STEAM DURING EXPANSION

Example number	1	2
Initial quality, parts of unity	0.540	0.950
Pressures, Lb. per Sq. In. Absolute		
Initial	145.0	145.0
Final	20.0	20.0
Value of n in Equation $PV^n = C$		
Adiabatic expansion, from Table 20	1.082	1.133
Actual expansion in engine tested, from Fig. 6	0.900	1.230
Volume of Steam Present, Cu. Ft. (Volume of water neglected)		
Initial	1.00	1.00
Final, adiabatic expansion	6.24	5.75
Final, curve of constant steam weight	6.44	6.44
Final, actual expansion in engine tested	9.04	5.00
Weight of Steam Present, Lb.		
Initial, plus water	0.594	0.338
Initial, steam only	0.321	0.321
Final, adiabatic expansion	0.311	0.287
Final, curve of constant steam weight	0.321	0.321
Final, actual expansion in engine tested	0.450	0.249
Quality of Steam at Final Pressure, Parts of Unity		
Adiabatic expansion	0.524	0.849
Curve of constant steam weight	0.540	0.950
Actual expansion in engine tested	0.757	0.736
Condensation or Re-evaporation, Parts of Unity		
Apparent re-evaporation	0.217	
Real re-evaporation	0.233	
Apparent condensation		0.214
Real condensation		0.113

53 Example 1 is a condition which obtains in the engine tested when using saturated steam at about 140 lb. gage pressure with a length of cut-off of about 3 per cent. All values of the qualities mentioned are portions of the total weight of mixture in parts of unity. The value of X_e is 0.540, a low value, yet one which often obtains in small engines. If this steam were expanded adiabatically

to the back pressure, 20 lb. absolute, the resulting quality would be 0.524, giving a condensation of 0.016. The expansion which actually takes place in the engine tested under these conditions results in a final quality of 0.757, showing that the apparent re-evaporation from the value of X_e has been 0.217. However, the steam mixture in expanding *did* expand adiabatically in order to give up heat to work, but the actual or what might be called the gross expansion, was changed in character by the re-evaporation of a large proportion of the water present, due to the return of heat from the cylinder walls and the consequent flashing into steam of part of the water when the pressure and the temperature were lowered. The real re-evaporation, measured by its effect upon adiabatic expansion, has been the difference between 0.757 and 0.524 or 0.233.

54 The actual expansion in this case has been the result of two factors which worked simultaneously: adiabatic expansion and the return of heat from the cylinder walls to the mixture. The first factor, adiabatic expansion, as already explained, is itself the result of two neutralizing or opposing conditions, i. e., the condensation of initially dry steam during expansion, and the relatively smaller amount of re-evaporation of water initially in the mixture due to the liberation of its excess of heat when the pressure and temperature were lowered. The net result of the two conditions of this adiabatic expansion, however, was a condensation. The second factor is the large amount of re-evaporation due to the return of heat from the surface of the cylinder walls to the condensed steam, amounting in Example 1 to 0.233, or, roughly, there has been re-evaporated during expansion $\frac{1}{4}$ of the entire weight of mixture present.

55 Example 2 shows conditions which obtain in the engine tested when served with steam superheated about 125 deg. fahr. at about 140 lb. gage pressure, with a length of cut-off of about 45 per cent. The value of X_e is 0.950, a very high value for this class of engine. The quality after adiabatic expansion would be 0.849, a condensation of 0.101. Where values of X_e are as high as 0.950, however, no re-evaporation takes place in practice, but condensation continues throughout expansion. After expansion in the engine tested, the quality would be 0.736, showing much greater condensation than that due to adiabatic expansion alone. The apparent condensation has been 0.214, but the real condensation, measured by its effect upon adiabatic expansion, has been 0.113.

56 The actual expansion in Example 2, as in Example 1, has been

the result of two factors: adiabatic expansion and the further abstraction of heat during the whole expansion by the cylinder walls. Heat is abstracted during expansion by the cylinder in the engine tested at all values of X_e above 0.85, thus giving values of n higher than the adiabatic value k.

57 The phenomena of condensation and re-evaporation during expansion are the causes of the relations existing between X_e and n in the cylinders of steam engines. The two examples given show values obtained in extreme cases which illustrate very well the effect of X_e upon the character of the expansion, and therefore upon the value of n, and show the range of values that n assumes in one engine due to a change in the value of X_e.

58 *The Relation of X_e and n.* The relation of X_e and n, for the same engine, has been shown by the author's tests to be very definite under the same conditions of pressure and speed. This relation, or dependence of n upon X_e, however, is not seriously affected by changes of pressure and speed within the limits of the tests.

59 The ordinary speeds of similar types of engines, 70 to 120 r.p.m., do not affect the relation seriously enough to be taken into account when examining such cases, because of the twofold character of speed in its influence upon the action of the cylinder walls.

60 The engine experimented upon was operated at 120 r.p.m., and had a stroke of 2 ft. Other engines of this class run at speeds as low as 70 r.p.m., but have strokes of 5 or 6 ft. Cylinder condensation is not dependent upon rotative speed alone, but is also influenced by the piston speed, as determined by the length of the stroke. On account of different lengths of stroke, different engines cannot be compared on the basis of rotative speed. Thus while the small engine tested has a rotative speed of 120 r.p.m., its piston speed is only 480 ft. per min. In large engines, while the rotative speed may be only 70 r.p.m., yet, with strokes of 6 ft., the piston speed is 840 ft. per min. What the small engine gained by higher rotative speed, the large engine made up in a measure by higher piston speed.

61 After taking into account the two parts of which speed is composed, it is found that the speeds of stationary engines of the type tested are in substantially the same range. On account of this fact, only the results of the tests run at 120 r.p.m. have been used in the applications made at present.

62 The relation of X_e and n has been found to be practically independent of cylinder size. This statement is true for non-jacketed

cylinders and for pressures in the range examined. This is shown in a general way in the following discussion for saturated steam.

63 It appears, therefore, that this method of approximating the value of X_c at cut-off from experimentally determined relations, and thus accounting for the initial condensation, is upon much surer ground than any method of computing condensation from the physical facts surrounding the case. It approaches the problem from the side where the phenomena occurring are practically independent of all the variables mentioned. This fact adds greatly to the accuracy obtained, and with the applications given in Table 4 shows that an accurate and reliable method of approximating the actual steam consumed from the indicator diagram alone has been devised.

64 This method is free from several objections to which tests are open. It measures the consumption in one revolution, and is, therefore, practically measuring a *rate* instead of a *quantity*. The only data needed for an approximation are one set of indicator diagrams, taken simultaneously, the constants of size and clearance, and the speed of the engine tested. No interruption of any kind in the routine of a plant is caused, and the expense incurred is not to be compared with that of an equally accurate test. The method is accurate enough for almost all purposes except guarantee tests subject to bonus and forfeit contracts. In the case of locomotives on the road, it is the *only* possible method of approximating the steam consumption of the main engines, due to the use of steam by the air-pump, train-heating system, blower, generator sets, safety valves, whistle, blow-off valves, and leaks. The same is true of marine engines, where many auxiliaries are supplied with steam from the same boilers, and exhaust into the same surface condensers. The method is especially useful for non-condensing engines, where the boiler-feed measurement method is the only practicable one. Steam consumption may be obtained as often as is desired instead of probably once in an engine's life.

65 *Limitations.* The relations of X_c and n given in Fig. 6 are applicable, however, only to non-jacketed cylinders exhausting at very close to atmospheric pressure. When the back pressure is raised to 30 lb. absolute, for instance, there is a new series of relations existing for the same initial pressure, due to a different temperature range in the cylinder and the consequent alteration of the phenomena occurring after cut-off. Steam jackets also alter the phenomena occurring after cut-off, and therefore have to be examined separately for the relations of X_c and n.

66 Since this method rests entirely on the indicator diagram, great care must be observed in taking these diagrams. The indicator itself must be an accurate instrument in the best possible condition. The indicator connections must be short and direct. An extensive investigation by W. F. M. Goss [1] shows that the long and indirect pipe connections materially alter the form and character of the expansion curves. A correct reducing motion, free from lost motion, must be used so as to reproduce the actual expansion. The arrangement of having one indicator at each end of the cylinder is always to be preferred.

67 The applications of this method must be made with judgment and care. If large leakage exists only an approximate solution can be obtained, as certain assumptions, treated in the section on leakage, have to be made. The various steps involved in the use of the method must be thoroughly comprehended to give satisfaction.

68 *Application of the Method.* The relations of X_e and n, as determined for various pressures at constant speed from the engine tested in the laboratory of the University of Illinois were plotted in the form of the chart shown in Fig. 6.

69 The next step was to examine, with certain restrictions, the tests of other engines, and to compare the relations of X_e and n with those given in Fig. 6. The restricting conditions imposed were: (*a*) that the tests should come from reliable sources; (*b*) that the data supplied should be complete enough for computing the quantities needed for comparison; (*c*) that the cylinders should be non-jacketed; (*d*) that the diagrams furnished should be representative of average conditions; (*e*) that the back pressure in the cylinder examined should be practically atmospheric; (*f*) that no large leaks should exist.

70 The values of X_e, n and p were first found from the set of diagrams to be examined. Next, the values of n and p were located in Fig. 6 and the corresponding value of X_e found, as obtained in the tests. The value of X_e obtained from the chart and that obtained from the test examined were compared, and the steam consumption, as computed by the value of X_e taken from the chart, was obtained.

71 The results of tests which fulfilled the conditions imposed are given in Table 4. Four distinct classes of engines were examined. These include simple Corliss, two-valve and four-valve types, the high pressure cylinders of compound engines, the intermediate pressure cylinders of triple expansion engines, high-speed and simple

[1] Trans. Am. Soc. M. E., vol. 17, p. 398.

locomotive engines. The sizes range from $10\frac{1}{2}$ in. by 12 in. to 34.2 in. by 60 in., and the speeds from 263 to 47.98 r.p.m.

72 The final results, given in columns 24, 29, 31 and 36 of Table 4 were averaged (analysis 201 excepted) and the averages are given as follows:

Average Difference from Test Results	Value of X_c, Per Cent	Actual Steam Consumption, Per Cent
Approximation from Chart—Fig. 6		
Irrespective of sign	3.06	3.72
Higher (+) or lower (−) than test results	−1.93	+2.50
Approximation from Equation $X_c = 1.245\,n - 0.576$		
Irrespective of sign	3.32	3.98
Higher (+) or lower (−) than test results	−1.32	+1.91

Analysis 201 shows an application, the conditions of speed and cut-off pressure of which are far outside of the limits examined. The values given were obtained by extrapolating as straight lines the lower portions of the curves of constant value of X_c in the chart of Fig. 6. Although the speed is only 27.66 r.p.m. and the cut-off pressure only 38.5 lb. absolute, the value of X_c by chart was determined as 0.470, while the value by test is 0.508, a difference of 7.5 per cent, based on the test value of X_c. This application is given to show that an extrapolation of the method to unusual conditions of speed and cut-off pressure does not lead to absurd results, although it is not nearly as accurate as the applications to speeds higher than 50 r.p.m.

73 The results of the applications made up to the present, with the restricting conditions imposed, tend to show that the steam consumption of engines may be approximated from the indicator diagram to within an average difference of less than 4 per cent from the test results. Individual examples, however, may show as much as 8 per cent difference in rare cases.

CONCLUSIONS REGARDING STEAM CONSUMPTION FROM DIAGRAMS

74 The following conclusions have been drawn from the results of the investigation, as applicable to non-jacketed steam cylinders in good physical condition exhausting at or near atmospheric pressure, and with the limitations imposed as already given.

a At a given initial pressure and speed of engine, there is a definite relation existing between X_e and n, in any one cylinder, which is practically independent of the cut-off position.

b This relation is practically independent of cylinder size and of engine speed; it is therefore applicable to other cylinders of the same type.

c By means of the experimentally determined relations of X_e and n, the value of X_e may be approximated from the average value of n obtained from the expansion curves of one set of indicator diagrams, taken simultaneously; therefore the actual weight of steam present in one revolution may be approximated.

d The actual steam consumption may be obtained by this method from the indicator diagram to well within an average of 4 per cent of the amount consumed as measured by test.

e This method has the following advantages not possessed by tests: it is more accurate than the average test, and is the only accurate method available for testing certain classes of engines; it virtually measures an instantaneous rate instead of an average quantity over a long time, and thus enables a large number of points to be obtained for a water-rate curve; it permits of making tests at frequent intervals instead of once in the engine's life; the expense is not to be compared with that of an equally accurate test; it involves no change in the routine of the plant tested.

THE LOGARITHMIC DIAGRAM APPLIED TO ALL ELASTIC MEDIA

75 *The Form of Expansion and Compression Curves from Practice.* About 300 indicator diagrams from 50 engines using steam, gas, air, and ammonia have been examined [1] to investigate the form and character of the expansion and compression curves. It may be stated as a result that the polytropic law $PV^n = C$ holds for all elastic media with certain exceptions which have been studied and the causes treated.

76 An example of two logarithmic diagrams showing how perfectly the law holds is given in Fig. 9. These diagrams were constructed from two indicator diagrams shown in Fig. 15 taken from the Corliss engine tested.

[1] See Appendix for examples and discussion.

77 *Mathematical Relations of the Law,* $PV^n = C$. The equation of the polytropic curve, $PV^n = C$, when plotted on rectangular cross-section paper, gives a curve depending for its form and position on the values of P, V, n, and C. When this curve is plotted on logarith-

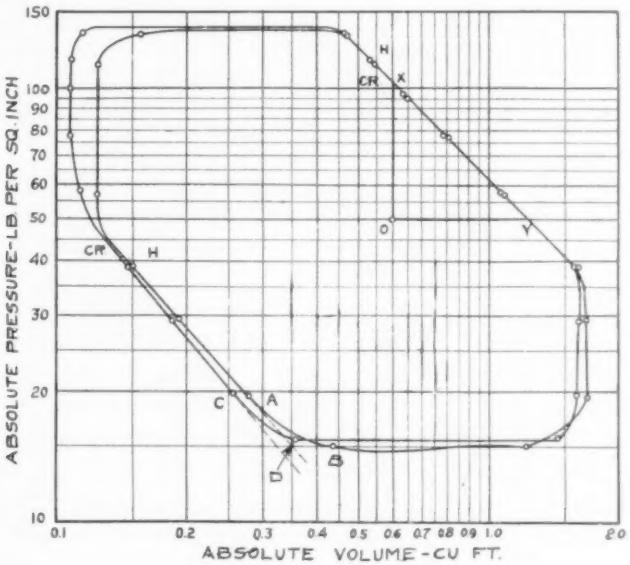

FIG. 9 LOGARITHMIC DIAGRAMS PLOTTED FROM FIG. 15

mic paper it becomes a straight line depending for its slope on the value of n and for its position upon the value of C. The relations for such curves are:

Given

$$PV^n = C$$

Taking the logarithm of both terms

$$\log P + n \log V = \log C$$

Transposing

$$\log P = -n \log V + \log C$$

This equation is of the form of the straight line

$$y = mx + b$$

where

$$y = \log P$$
$$m = -n$$
$$x = \log V$$
$$b = \log C$$

Thus $m = -n$, the slope, or measure of inclination of the line to the axis log V. In Fig. 9, for example, at a point X on the line

$$\log P = -n \log V + \log C$$

draw OX parallel to the axis log P, and draw OY parallel to the axis log V. The slope of the line will be the value of the ratio $\frac{OX}{-OY} = -n$.

OY is negative, being measured to the left, giving n its negative sign.

78 *Use of the Logarithmic Diagram.* The logarithmic diagram forms the basis of the methods of analyzing the cylinder performance of reciprocating engines which are developed in detail in the following pages.

79 These methods apply only, however, to the logarithmic diagrams derived from the cylinders of reciprocating engines using an elastic fluid for the working medium and having, as a part of the cycle of operation, an expansion, a compression, or both.

80 The figures of one set of indicator diagrams and the corresponding set of logarithmic diagrams are numbered the same, but the letters a and b are used in addition to the figure number to denote the indicator and logarithmic diagrams, respectively.

RATIONAL METHOD OF APPROXIMATING CLEARANCE

81 In the cases of the great majority of the PV-diagrams which were examined,[1] the expansion and compression curves became straight lines in the logarithmic diagram, showing that the law $PV^n = C$ was applicable, or in other words, that n was a constant for one curve. The clearances furnished with the diagrams examined had been carefully found by the displacement method.

82 It was desired to see what forms the lines assumed when the clearance was taken larger or smaller than the measured quantity. The diagram shown in Fig. 10-a, taken from a 42 in. by 60 in. gas engine, was used for this purpose. The true clearance, measured as 18.0 per cent, was used in the full logarithmic diagram of Fig. 10-b. Trials were made with clearances assumed as 14.0, 16.0, 20.0, and 22.0 per cent of the piston displacement. With the true clearance of 18.0 per cent, the curves became almost perfectly straight lines, while with the values of clearance less than 18.0 per cent, it is seen that the lines become bent to the left, and with values of over 18.0 per cent the lines become bent to the right. Hence the straight

[1] See Appendix for examples and discussion.

line for the value of 18.0 per cent is the transition between the family of curves bending to the left, representing a clearance smaller than the real value, and the family of curves bending to the right, representing a clearance larger than the real value.

83 The practical significance of this fact is that there is now available a rational method of approximating the clearance of any

FIG. 10-*a*

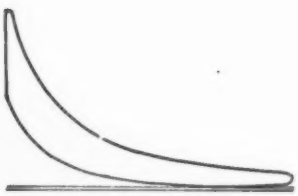

FIG. 11-*a*

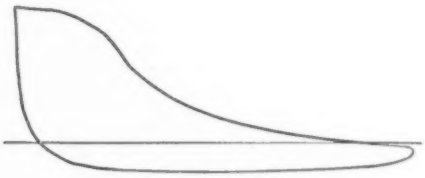

FIG. 13-*a*

FIG. 14-*a*

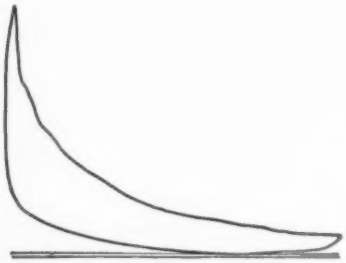

FIG. 12-*a*

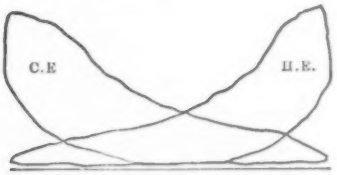

FIG. 16-*a*

DIAGRAMS FROM GAS ENGINES, CORLISS ENGINE, COMPRESSOR AND HIGH-SPEED STEAM ENGINE

cylinder using an elastic medium, which has, as a part of the cycle of operation, an expansion or a compression. This method is based on the fact, already mentioned, that in practice all elastic media, except under certain exceptional conditions, obey substantially the law $PV^n = C$, when subject to change of state, and therefore become straight lines in the logarithmic diagram.

84 *Graphical Method of Approximating Clearance.* The graphical method of approximating clearance requires only the scale of the

indicator spring to be known, and the atmospheric line to be drawn, in order to locate the zero line of pressure. The exact order of procedure necessary to make a trial, and the degree of accuracy obtained in any given case, is shown in detail in Par. 93 for a $25\frac{5}{8}$ in. by $37\frac{5}{8}$ in. gas engine. All that is necessary is to assume different values of clearance, and plot the logarithmic diagram for each assumed value. The straight line position of the curves is found by trial and error, to lie between the two diverging families of curves representing too small and too large clearance.

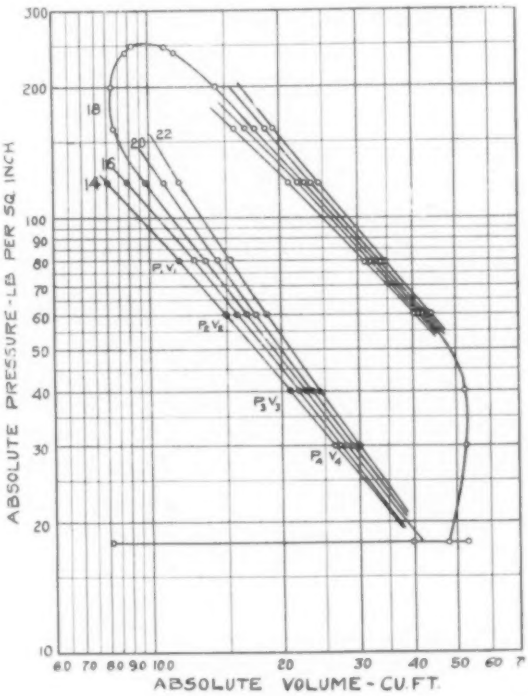

Fig. 10-b Tod 42 in. by 60 in. Gas Engine, Blast Furnace Gas

85 It follows also from the curves shown by Fig. 10-b that the clearance being known, the scale of the spring used may be obtained in the same manner if the atmospheric line is given.

86 *Mathematical Method of Approximating Clearance.* The results obtained from the graphical method of trial and error may also be accomplished by the use of the purely mathematical process upon which the method depends.

87 The law $PV^n=C$ holds, thus n is a constant for any part of the curve. When the wrong clearance is used, the law $PV^n=C$ does not hold, and n varies from point to point. In the graphical method, trials of various values of clearance are made, until the curve becomes approximately a straight line; this resultant straight line is, therefore, the law $PV^n=C$, in which n is a constant for all parts of the curve. The one condition necessary to be fulfilled, therefore, is that n be constant for all parts of the curve, but not of any particular value.

88 To illustrate the use of the mathematical method in Fig. 10-b, let us assume several points, as P_1V_1, P_2V_2, P_3V_3, and P_4V_4 at various intervals on one of the curves, as on the compression curve at the clearance value 14.0 per cent. It is desirable for convenience to locate the points at about equal intervals, as shown. The law $PV^n=C$ is assumed to hold. Then, for two points, P_1V_1 and P_2V_2, called group *a*, we have

$$P_1V_1^{\,n}=C$$
$$P_2V_2^{\,n}=C$$

Equating these, we obtain

$$P_2V_2^{\,n}=P_1V_1^{\,n}$$

Transposing and dividing

$$\left(\frac{V_2}{V_1}\right)^n=\frac{P_1}{P_2}$$

Taking the logarithm of both sides

$$n\,(\log V_2-\log V_1)=\log P_1-\log P_2$$

Whence

$$n_a=\frac{\log P_1-\log P_2}{\log V_2-\log V_1}\quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[1]$$

In the same manner for the points P_3V_3 and P_4V_4, called group *b*, is obtained

$$n_b=\frac{\log P_3-\log P_4}{\log V_4-\log V_3}\quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[2]$$

For the correct value of clearance, the following condition must be fulfilled by trial and error

$$n_a=n_b\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[3]$$

89 The values of the logarithms of the coördinates of all the points are then found, and the values of n_a and n_b computed. If the points are located in the order shown, then with too small a clearance, n_a is lower in value than n_b. A larger value of clearance is then assumed, the operation being merely to add a constant number

to the values of V_1, V_2, V_3, and V_4. The process is repeated until the value of n_a becomes practically equal to n_b. When the clearance is assumed too large, n_a becomes higher in value than n_b, indicating that the true value has been passed.

90 The trial by the mathematical method is neither as accurate nor as short as the graphical method. It is not as accurate because the points assumed may not be representative. When this is the case, the graphical method allows judgment to be exercised in selecting the straight-line position, thereby eliminating irregularity of points.

91 The question arises as to whether the form of the lines due to wrong clearance can be distinguished from the form due to leakage or to "hooks," on the logarithmic diagram. This case is treated in Par. 126.

92 The curve of the PV-diagram nearest the clearance space, or the compression curve in Fig. 10-b, is generally the better guide in the graphical trials. This is well shown in Fig. 10-b. A given difference in the values of clearance used for trial causes more horizontal variation in the position of the compression curve than in the expansion curve. This fact allows closer locations of the straight-line transition region to be made from the compression curve than from the expansion curve.

93 *Examples.* It was desired to determine the clearance of the diagram shown in Fig. 11-a. From general knowledge of this class of engines a trial by the graphical method was made in Fig. 11-b with the clearance assumed as 12.4 per cent, a value purposely assumed as being too small. This value is seen, by the bending of both curves to the left, to be much too small. Trials were, therefore, made with the clearance assumed as 13.8, 15.1, and 16.0 per cent of the piston displacement. The values of 15.1 per cent gave practically straight lines for both the expansion and compression curves, while the value of 16.0 per cent shows that the lines have begun to bend to the right, indicating too large a clearance. By inspection, it will be seen that the region of fairly straight lines may be located as lying between the values of about 14.5 per cent to 15.5 per cent. The clearance is, therefore, selected as 15.0 per cent, a value which may be high or low by not more than 4 per cent in this case. This clearance value, 15.0 per cent, is a common value for engines of this class.

94 The graphical method is more accurate for large clearances, measured in per cent of the piston displacement, than for small ones.

The closeness of location of the straight-line region, lying between the two families of diverging curves, will be found to be within 5 per cent to 10 per cent of the clearance volume, for values of clearance

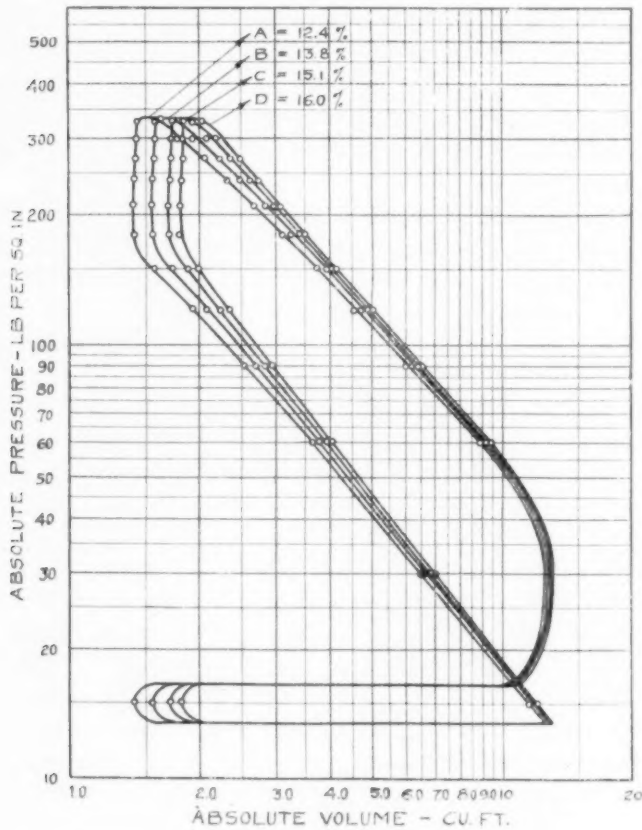

Fig. 11-b Koerting 4-Cycle 25¾ in. by 37¾ in. Producer Gas Engine

between 20 per cent and 2 per cent, respectively, of the piston displacement.

RATIONAL METHOD OF LOCATING THE STROKE POSITION OF CYCLIC EVENTS

95 It is often desirable to know at what part of the stroke the cyclic events occur. This knowledge can be best obtained from the PV-diagram. For ordinary purposes, these events can be closely located by inspection on the PV-diagrams themselves, in most cases; thus, on a diagram from a Corliss engine, cut-offs may generally be

located to within $\frac{1}{16}$ in., measured along the length of the diagram.

96 The actual beginning of true compression, however, can *never* be accurately located on the PV-diagram. True compression, unaffected by leakage, begins after the exhaust valve, in closing, has acquired enough seal to prevent leakage. The point of the begin-

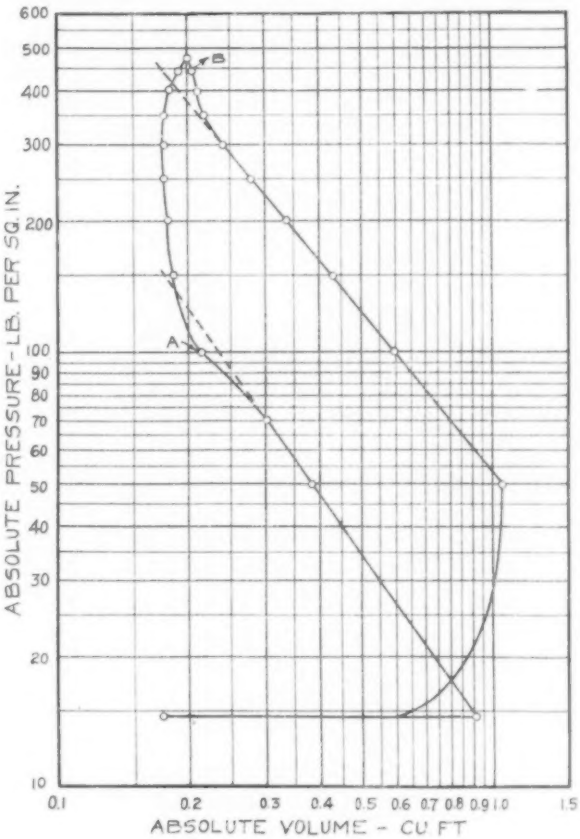

FIG. 12-b GAS ENGINE 10 IN. BY 19 IN. FOR ILLUMINATING GAS

ning of true compression is generally at least 5 lb. above the back pressure. The point at which leakage ceases cannot be located on the PV-diagram because the curve of true compression, and the curve during the time the valve has insufficient seal, are of the same direction of curvature, and are not reverse curves as in the general case of admission and expansion.

97 The fact that expansion and compression of a constant weight of medium takes place[1] according to the law, $PV^n = C$, thus becoming straight lines in the logarithmic diagram, enables us to locate cyclic events very closely, even in cases where they cannot be detected at all in the PV-diagram.

98 An example is shown in Fig. 21-a, containing locomotive

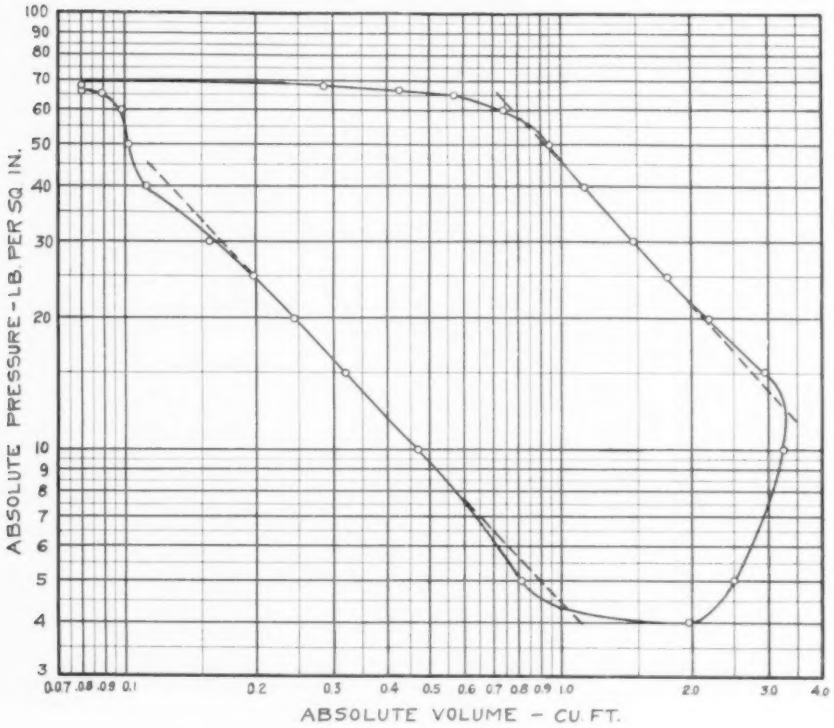

FIG. 13-b CORLISS ENGINE 14 IN. BY 35 IN.

PV-diagrams taken at short cut-off and high speed. The events of cut-off, release, compression, and lead are very difficult to locate on such diagrams. These events are located on the logarithmic diagram in Fig. 21-b by noting when the expansion and compression curves become straight, indicating a constant weight of steam mixture.

99 A sufficient number of points are plotted to show clearly the direction of the diagram near the events desired. Thus these events,

[1]See Appendix for examples and discussion.

even though obscure in the PV-diagram, may be located to well within about $\frac{1}{16}$ in. in the logarithmic diagram, this length being equivalent to about $\frac{1}{32}$ in. when retransferred to the PV-diagram itself.

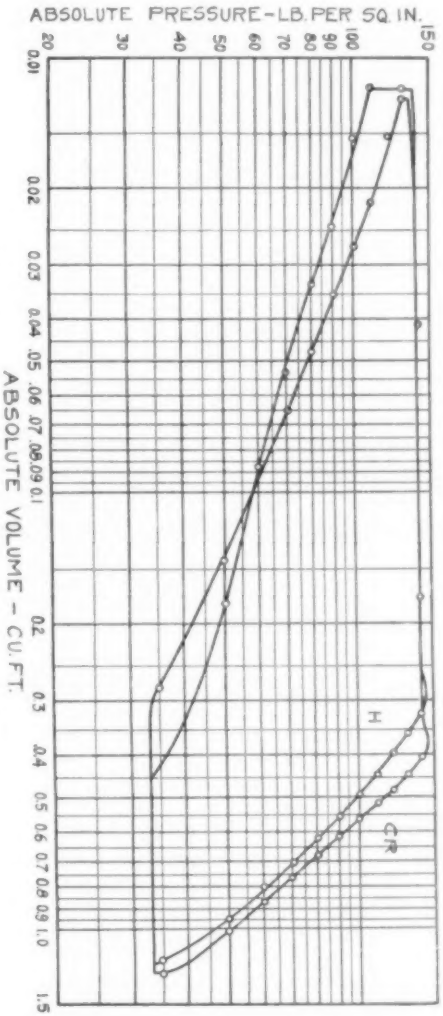

FIG. 14-b AMMONIA COMPRESSOR 11¼ IN. BY 22 IN.

100 The use of this method has one great advantage in that it largely eliminates the variable element of personal judgment. It is a common occurrence to see PV-diagrams where two persons have

located an event such as cut-off, $\frac{1}{8}$ in. apart, each location being the best judgment of the person doing the work. The logarithmic diagram will at all times give closer locations of events for these reasons than will the PV-diagrams.

101 The method also allows the point of true compression to be located, the location of which is practically impossible in the PV-diagram.

RATIONAL METHOD OF DETECTING LEAKAGE

102 The law $PV^n = C$ is applicable only to cases where the weight of the working medium remains practically constant during any

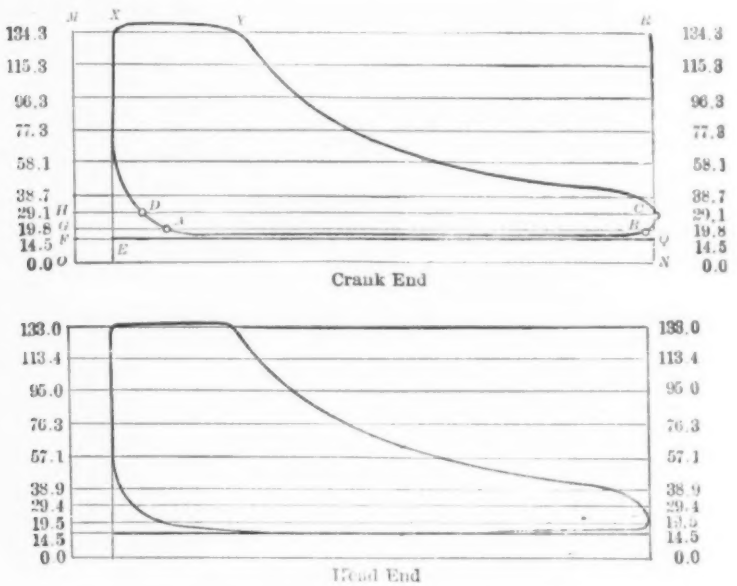

FIG. 15 DIAGRAMS TAKEN FROM TEST 30, TABLE 1, 12 IN. BY 24 IN. CORLISS ENGINE

THIS AND THE SUBSEQUENT DIAGRAMS ARE REFERRED TO IN THE APPENDIX AND ARE GIVEN TO ILLUSTRATE TYPICAL LOGARITHMIC DIAGRAMS

expansion or compression. When this weight changes materially, either by leakage into, or out from, the cylinder containing the medium, the resulting expansion or compression no longer obeys the law, and it becomes a curve on logarithmic cross-section paper. This fact is very clearly shown in the curves of the logarithmic diagram derived from cylinders in which large leaks were known to exist.

103 *Examples of Known Leakage.* The first case, shown in Fig.

12-b, occurred in a 10 in. by 19 in. gas engine, intended for producer gas, but using illuminating gas at high compression. The piston, a single-acting trunk type, allowed a large leak, clearly detected by the noise of escaping gas, at the beginning of the combustion stroke. Both the compression and expansion curves show the effect of this leak in a clear manner when transferred to the logarithmic form. After that portion of the stroke was reached where no sound of leakage was heard, the two curves became straight lines. This in-

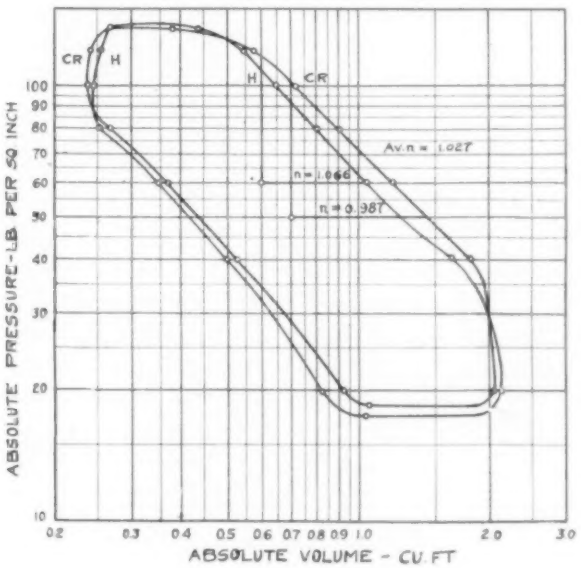

Fig. 16-b Ide 16 in. by 16 in. High-Speed Steam Engine

dicated very clearly that the effect of leakage, if appreciable, may be detected in the form of the curves of the logarithmic diagram.

104 The second case, shown in Fig. 13-b, is from a 14 in. by 35 in. Corliss engine. The knowledge of the leaky condition of the piston and valves came from the engineer in charge.

105 The expansion and compression lines indicate by their form at the upper ends, a large leak from the cylinder, or through the exhaust valve. The lines also show, by the rising of the curves at the lower ends, a considerable addition to the steam in the cylinder during expansion and compression. This steam could come only from a large leak in the steam valve. The seven other diagrams taken

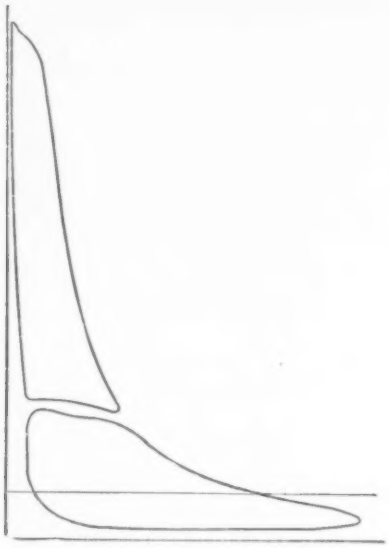

FIG. 17-a DIAGRAM FROM RICE AND SARGENT 16 IN. BY 28 IN. BY 42 IN. SUPERHEATED STEAM ENGINE

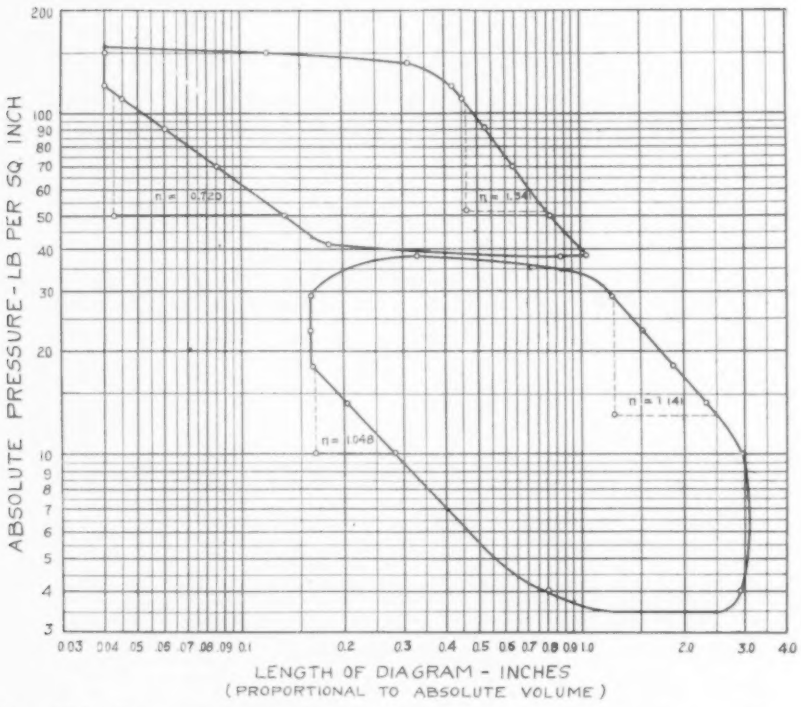

FIG. 17-b RICE AND SARGENT 16 IN. BY 28 IN. BY 42 IN. STEAM ENGINE USING HIGHLY SUPERHEATED STEAM

from this same engine all showed the effect of leakage in a similar manner.

106 The third case of known leakage, shown in Fig. 14-b, is from an $11\frac{1}{4}$ in. by 22 in. double-acting ammonia compressor. This cylinder was known to be in a very bad condition as regards wear and leakage of piston and valves. The re-expansion curves, by the enormous amount of re-expansion shown, indicate large leakage into the cylinder during this operation. The lower part of the compression curves, by rising, indicates leakage into the cylinder, either past the piston or through the discharge valves. The upper part of these curves indicates leakage from the cylinder, either past the piston or through the suction valves.

107 These three examples show abnormal conditions which are comparatively rare. Very smooth curves may be obtained in the PV-diagram even if there is large leakage taking place. This is seen by referring to Fig. 13-a, both the expansion and compression curves being fairly regular. The logarithmic diagram, however, shows clearly, in connection with the discussion and the examples shown, that large leakage of two kinds was taking place during expansion and compression. Leakage which occurs during admission or during exhaust has no effect upon the lines of the diagram as the weight of the medium is continually changing.

108 *Method of Detecting Leakage.* When leakage occurs in a cylinder, it is seldom found that only one source of leakage exists. Leakage is usually the result of wear, which affects most of the possible sources of leakage in about an equal proportion. As a result, several leaks are generally affecting the curves. This is the case in Fig. 13-b and 14-b. In Fig. 13-b leakage was taking place both into and out from the cylinders.

109 In discussing leakage, it must be kept in mind that difference in pressure between two regions is the cause of this phenomenon. In the steam engine, there are three pressures which must be considered, i. e., the pressure in the steam chest, in the cylinder at the point discussed, and in the exhaust passage. Leakage, being due to difference of pressure, becomes material only when this difference becomes considerable. Thus leakage into, or out from a steam cylinder has been found to occur in most cases only when the pressure difference is over about 20 lb. In Fig. 13-b, the leakage into the cylinder, shown by the lower parts of the lines, begins to occur at about 25 lb. absolute, or 35 lb. lower than the pressure at admission. The leakage out from the cylinder, shown by the upper parts of the

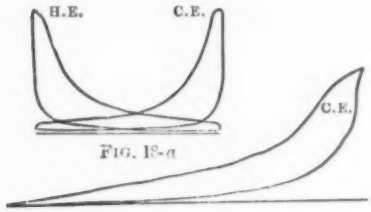

FIG. 18-*a*

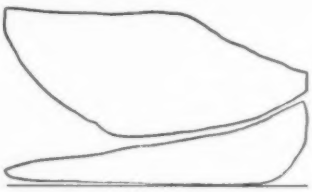

FIG. 19-*a*

FIG. 21-*a*

FIG. 22-*a*

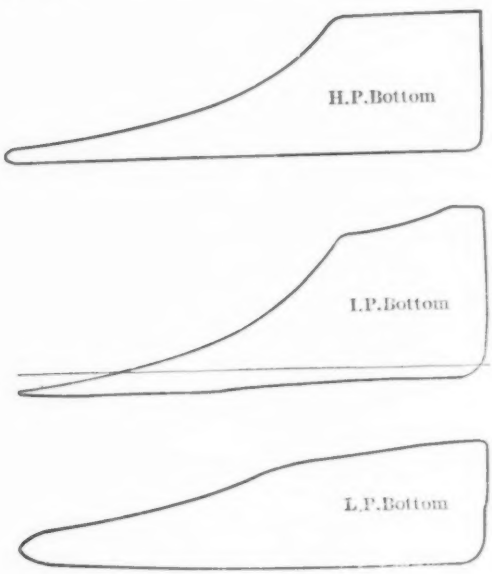

FIG. 20-*a*

DIAGRAMS FROM STUMPF ENGINE, SINGLE-ACTING WESTINGHOUSE, LOCOMOTIVE, DIESEL ENGINE AND PUMPING ENGINE

lines, ceases to occur at a pressure of about 40 lb. absolute for the expansion curve, and begins to occur at about 25 lb. absolute for the compression curve. The difference of pressure between the steam in the cylinder and that in the exhaust passage is about 35 lb. in the first case, and about 20 lb. in the second case.

110 This fact, founded on many diagrams analyzed enables us to

divide the expansion and compression lines roughly into three equal parts on the logarithmic diagram (when these lines extend from the initial pressure nearly to the back pressure): (*a*) the upper third, influenced by leaks *out from* the cylinder; (*b*) the middle third, practically *uninfluenced* by leakage; (*c*) the lower third, influenced by

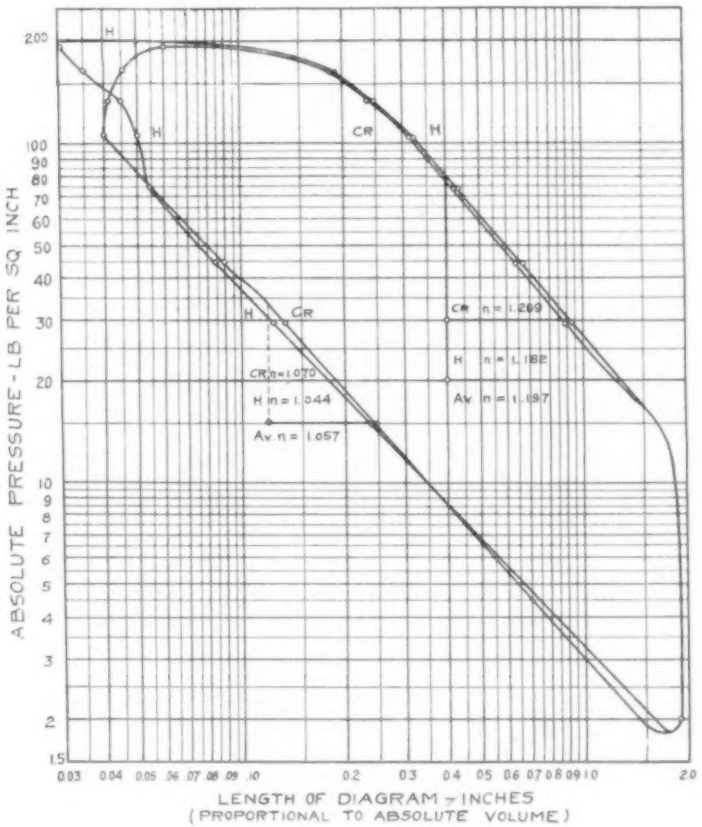

FIG. 18-b STUMPF 23½ IN. BY 31½ IN. UNI-DIRECTIONAL FLOW STEAM ENGINE USING SUPERHEATED STEAM

leakage *into* the cylinder. Thus fairly reliable values of n, free from the effect of leakage, may be obtained from the middle third of the lines.

111 Returning to Fig. 13-*b*, both the lines indicate leaks out from the cylinder. This can occur either past the piston or through the exhaust valves. The piston generally becomes leaky sooner than

Corliss exhaust valves, and, in this particular engine, one of the piston rings was found to be broken upon examination. When diagrams from both ends of the cylinder are available, piston leakage causes nearly an equal effect on the expansion curves of both ends. The leakage into the cylinder can come from only one source which can influence the curves, i. e., the steam chest. The effect of this leak is seen in both lines in the lower thirds.

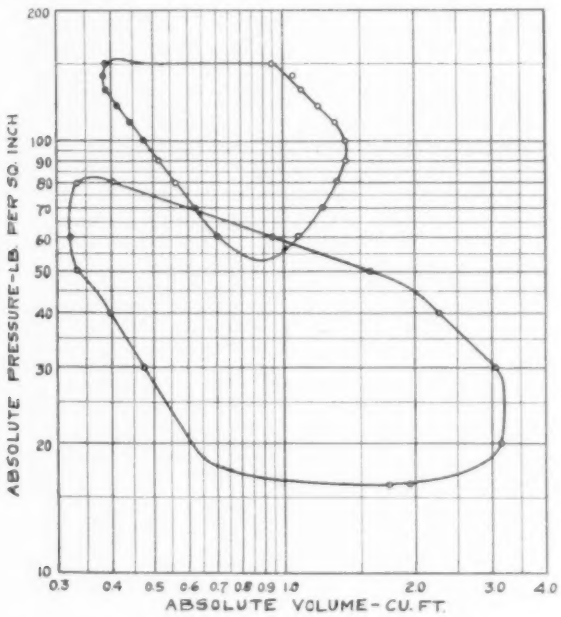

FIG. 19-b WESTINGHOUSE SINGLE-ACTING 13 IN. BY 22 IN. BY 13 IN. STEAM ENGINE

112 In Fig. 14-b, both forms of leakage are shown in the ammonia compression curves. The lower thirds show leakage into the cylinder, either through the discharge valves or past the piston. The upper thirds of the lines show large leakage from the cylinder, caused by the condition of either the suction valves or the piston. Ordinarily, it is not possible to distinguish between two leaks occurring in the same third of the curves.

113 *Approximation of the Volume Which Leaked.* Fig. 12-b is an example where only one kind of leakage is present. Here, the piston alone leaked badly at the commencement of the stroke. The effect of this leak is seen in the upper third of both lines.

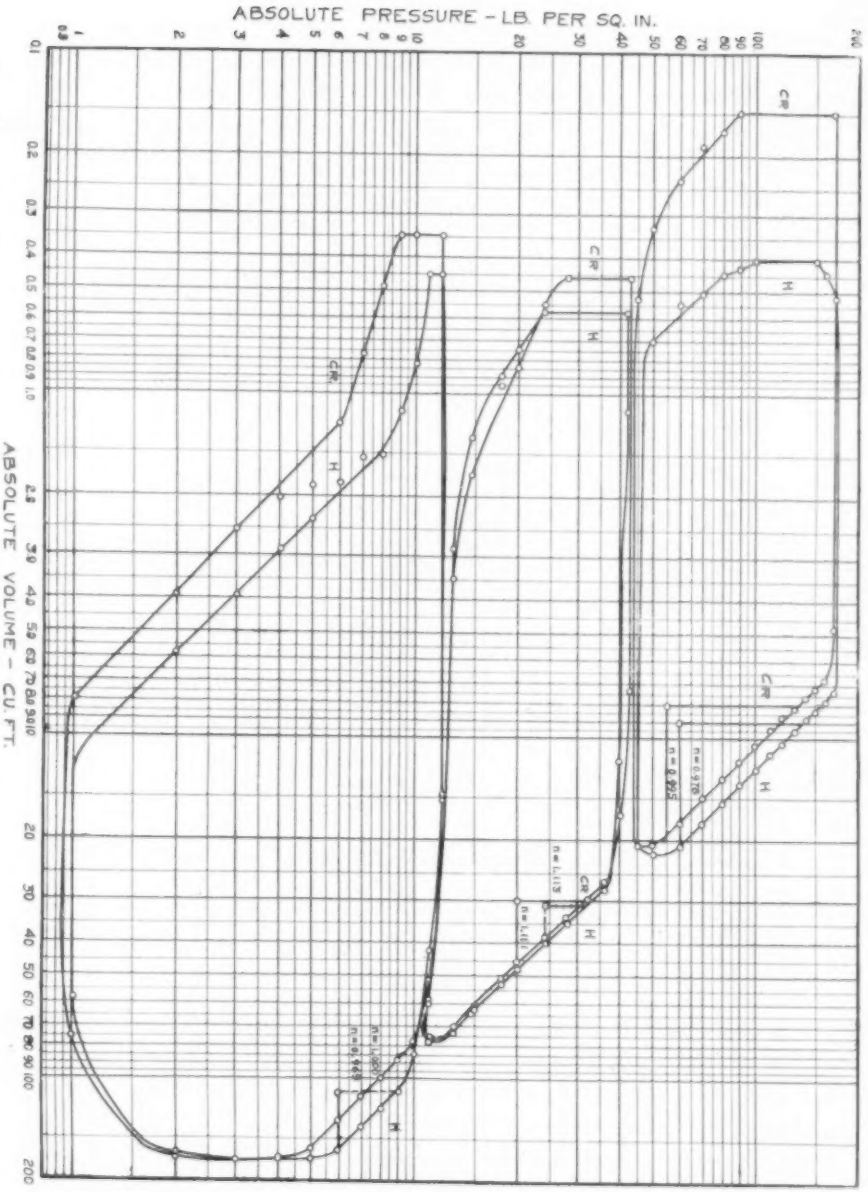

FIG. 20-b ALLIS-CHALMERS PUMPING ENGINE 28 IN. BY 54 IN. BY 80 IN. BY 60 IN. SATURATED STEAM

114 When only one kind of leakage exists it is possible to compute with fair accuracy the volume of leakage taking place during expansion or compression. The lines are extended, as shown in Fig. 12-b, giving the lines of constant weights of the medium. The volume of gas that had leaked during compression, up to 100 lb. absolute pressure, is then seen to be 0.014 cu. ft., or 6.3 per cent of the volume remaining. The volume of gas, measured at the pressure of 450 lb. absolute, that leaked after combustion during expansion,

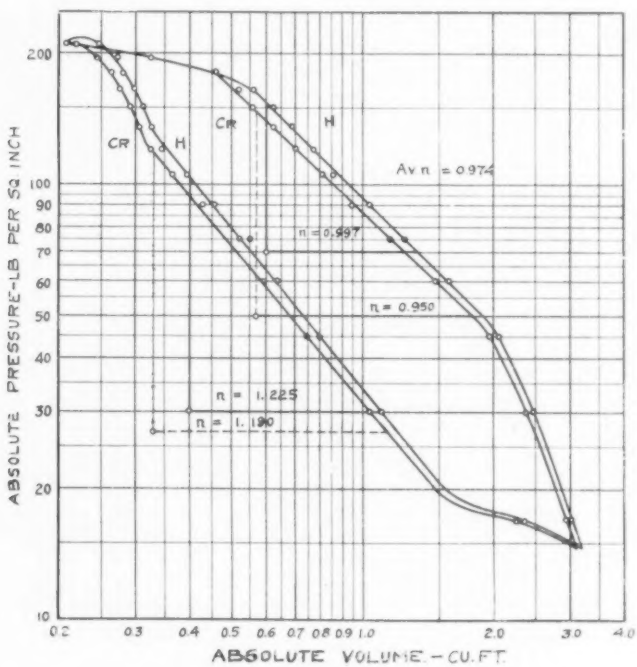

FIG. 21-b PURDUE 16 IN. BY 24 IN. SUPERHEATED STEAM LOCOMOTIVE.

is seen to be 0.032 cu. ft., or 18.7 per cent of the volume remaining after the leakage stopped.

115 The leakage that took place during combustion at the end of the stroke cannot be computed, but it can be estimated by making the assumption that this leakage was proportional to the mean rate of leakage shown by the two curves, and that its duration was the time interval occurring between the point *A* and the point *B*.

116 The important result that is attained by this method is not,

however, the approximation of leakage, but the knowledge that it is taking place, so that it can be located and stopped.

117 *The Use of the Method in Testing for Maximum Economy.* Many engines are sold and their prices fixed on the basis of their test

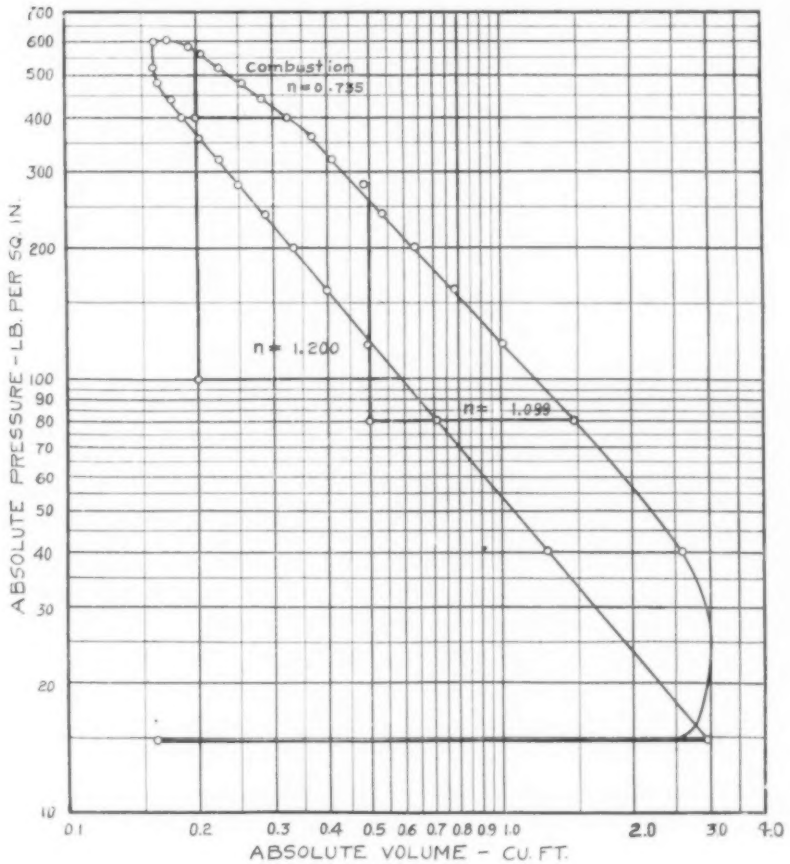

FIG. 22-b DIESEL 16 IN. BY 24 IN. OIL ENGINE, CRUDE OIL

performances. The importance to the manufacturer of being able to eliminate leaks during this test does not have to be emphasized. Does the engineer in charge of the test know whether the engine is tight under regular operating conditions?

118 All of our present knowledge of leakage is an inference drawn from the leakage "standing." Nobody knows whether an engine

that is tight "standing" leaks when in operation, or vice versa.

119 This method should be applied to all engines about to undergo any test where maximum economy is the object desired.

120 The results of an analysis made in connection with this investigation by means of logarithmic diagrams of a large number of PV-diagrams, indicate that the majority of engines, in good con-

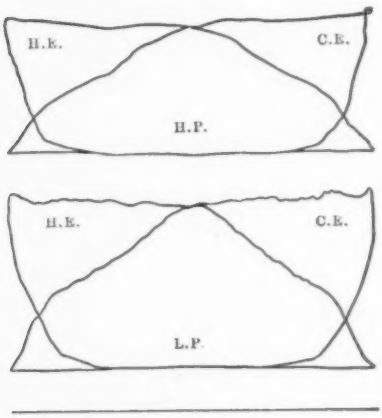

FIG. 23-a

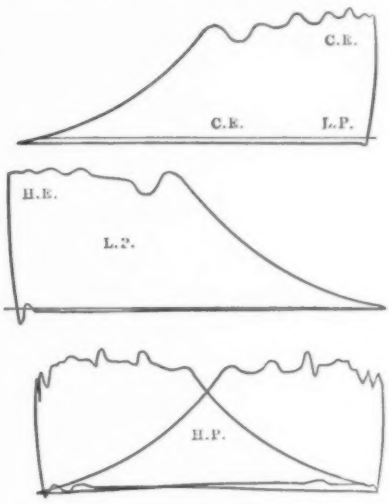

FIG. 24-a

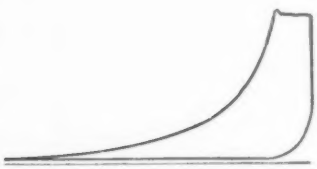

FIG. 25-a

FIG. 26-a

DIAGRAMS FROM COMPOUND, TWO-STAGE AIR COMPRESSOR, SINGLE-ACTING AMMONIA COMPRESSOR AND CORLISS ENGINE WHEN STARTING

dition, are practically tight as regards leakage into or out from the cylinder.

121 Most leakage is the result of wear and tear due to long and hard use. After the wear and tear has become marked, it is the custom to rebore the cylinder and to resurface the valves and valve-seats. Some railroad companies overhaul the cylinders and valves of locomotive cylinders at regular intervals of, for instance, 150,000 miles of travel. Some cylinders in stationary plants are rebored at equal time intervals of some four or five years each.

122 As a pure question of economy, other things not considered, general repairs of cylinder and valves should take place when the extra annual cost of fuel and water due to leakage equals the annual interest on the money necessary for general repairs.

123 It is apparent that the existing methods of determining when general repairs are necessary are in a very haphazard state as regards economy, and that the personal judgment of the man making the decision may be liable to great variation.

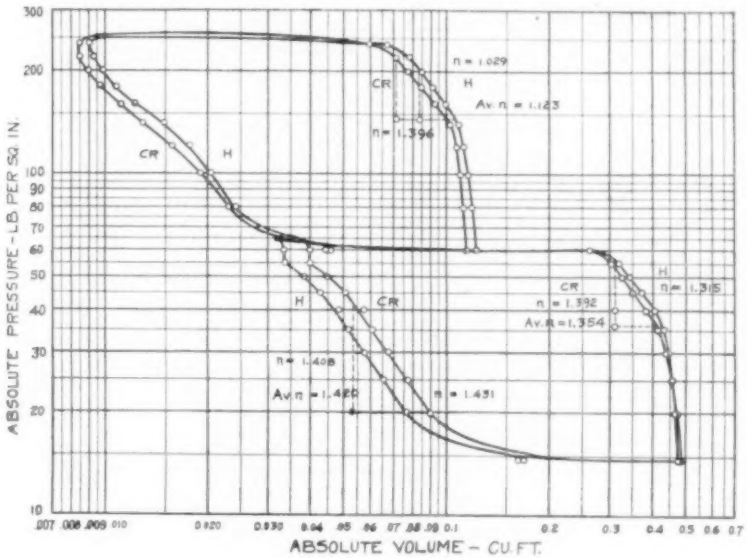

FIG. 23-b PORTER AIR LOCOMOTIVE-COMPOUND 5 IN. BY 10 IN. BY 10 IN.

124 The method of detecting leakage from the logarithmic diagram offers a more rational solution of this important question.

INTERPRETATION OF THE LOGARITHMIC DIAGRAM

125 In the discussion upon the effect of wrong clearance, leakage, and excessive condensation [1] upon the lines of the logarithmic diagram, it was assumed, for the sake of clearness, that only one of these effects existed at one time. The examples were selected so as to illustrate only one of these effects in each case.

126 Cases occur where several of these effects exist at one time in the same diagram. The separation of one effect from another is not an exact process. However, the character of the curves showing

[1] Discussed in Appendix.

excessive condensation, wrong clearance, and leakage are quite different. For instance, wrong clearance affects the lines throughout

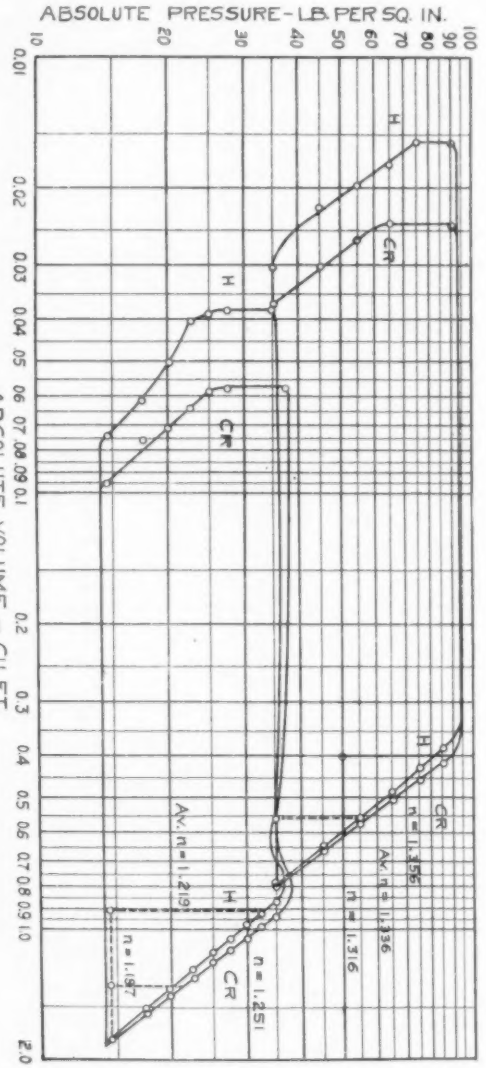

FIG. 24-b INGERSOLL-SERGEANT 12½ IN. BY 18½ IN. BY 12 IN. TWO-STAGE AIR COMPRESSOR

their length. Excessive condensation, in the cases of the steam diagrams examined, always affects only the upper parts of the curves. Leakage, as has been mentioned, affects materially only the upper and lower thirds of the lines, where these lines extend

from the initial pressure to nearly the back pressure. When excessive condensation and large leakage exist together, no close approximation of the clearance can be made.

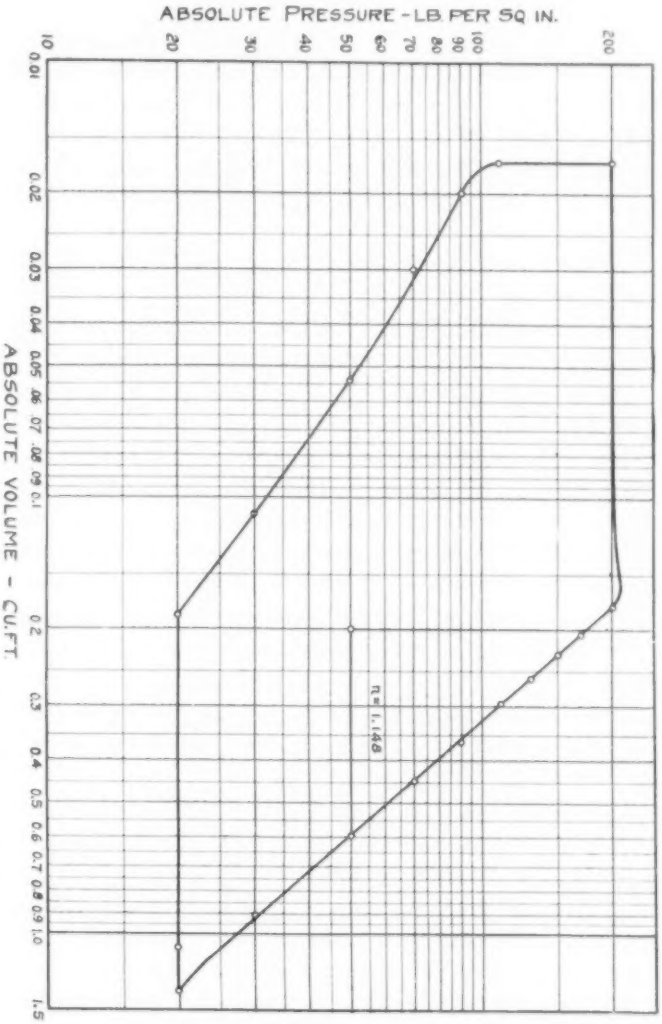

FIG. 25-b YORK SINGLE-ACTING 12½ IN. BY 18 IN. AMMONIA COMPRESSOR

127 An adequate treatment of the segregation of these various effects, when found together, is beyond the scope of this paper. The treatment is long and complicated. It has been found, however, that experience in the use of logarithmic diagrams enables one to sepa-

rate these effects qualitatively, in some cases, from the form of the expansion and compression curves.

COMMON ERRORS MADE IN ANALYZING STEAM INDICATOR DIAGRAMS

128 *The Use of the Equilateral Hyperbola as a Standard of Comparison.* The values of n for the expansion curves of steam indicator diagrams are not closely constant, but are subject to a very wide range of variation. The extent of this variation may be seen from the examples given in the logs of the tests made, and in the Appendix. The range of variation found in the present investigation is from 0.70 to 1.34.

129 The range of values in the engine tested was from 0.835 to 1.234. The average values were 0.947 for the tests run with saturated steam, 1.056 for the tests run with superheated steam, and 1.004 for all tests. The values of n for most engines of ordinary size using saturated steam at normal cut-off is between 0.95 and 1.05, while for superheated steam the range is usually from 1.00 to 1.30. For saturated steam the value of $n=1.0$ is about an average value. The only meaning that the average value of $n=1$ ever possessed is that the average value of X_o, in the class of engines examined, lies in the range between 0.60 and 0.70.

130 The law of Boyle or Mariotte, or the law of isothermal expansion of a perfect gas, has no bearing of any kind whatsoever on the expansion of steam in a cylinder. The equilateral hyperbola sometimes occurring in steam cylinders is only a special case of expansion according to the polytropic law $PV^n=C$, while Boyle's law is another special case which never occurs in steam engine practice.

131 Because of the agreement in form between Boyle's law and the equilateral hyperbola (the special case of the law, $PV^n=C$, where $n=1.0$), this latter curve has been called the ideal or theoretical curve of expansion to which curves in practice are supposed to approach as a measure of practical perfection in the use of steam. The equilateral hyperbola is in no sense whatsoever an ideal or theoretical curve, and its use for the purposes of comparison is an empirical or arbitrary convention only. It should be called the conventional expansion. It has even been contended that because an expansion curve did not coincide with the equilateral hyperbola, some grave fault exists in the engine. A value of n may be as low as 0.60 with no graver fault than very excessive initial condensation, while a

value of 1.35 may be found from no graver fault than that of using steam superheated about 250 deg. fahr.

132 The only rational use of applying the equilateral hyperbola to steam PV-diagrams is to act as a guide to see whether n is greater or less than 1.0. If the actual curve is not close to this hyperbola, if no faults exist, and if the cylinder is non-jacketed, then this fact means that the value of X_e for the case examined is less than about 0.60 or greater than about 0.70. The assumption that $n=1.0$ as a standard of expansion is equivalent to assuming that the value of X_e is standard at about 0.65. No engineer would seriously propose that the value of X_e of 0.65 should be selected as a standard of economy.

133 The elaborate theory of analysis built on the assumption that $n=1.0$ is the natural result of the use of averages in any art where the actual facts have never been investigated.

134 The use of the equilateral hyperbola to predict the form of PV-diagrams for the purposes of design is satisfactory in the case of ordinary sizes of engines using saturated steam. When steam superheated over 100 deg. fahr. is used, the value of n should be assumed at between 1.10 and 1.25. The high values of n obtained with highly superheated steam in large engines alter materially the division of work and the tangential forces acting from those obtained when n is assumed to be 1.0. This fact should be considered in the design of engines to use superheated steam.

135 The use of the equilateral hyperbola to obtain the ratio known as the "diagram factor" has no rational basis, but its use for this purpose gives results which are valuable for the purpose of design.

136 *The Graphical Method of Approximating the Clearance.* If n has the value 1.0 on a PV-diagram, the clearance may be found by locating the zero of volume on the zero line of pressure. This process is performed graphically by reversing the method used in constructing the equilateral hyperbola.

137 In actual expansions, however, n is almost never exactly equal to 1.0, but is greater or less as already explained. The accuracy of the result by this method is dependent on how close the value of n approximated 1.0. The clearance obtained by this method may be as much as 100 per cent larger or smaller than the actual clearance volume in ordinary cases, while errors of 25 per cent and 50 per cent are very common. Where errors of this size are possible, the method is of n o use for important work.

138 A rational method of approximating the clearance cannot be based upon the assumption that $n=1.0$, but only upon the fact that it be of a constant value, the value itself being immaterial.

139 *Combined Steam Indicator Diagrams.* Very little of value is obtained from the combined PV-diagrams of steam engines, except the measure of the diagram factor for the purpose of design.

140 One of the uses that has been made of the combined diagram is to see whether continuity of expansion exists. It has been assumed by various writers that continuity of expansion should exist. A study of the relations of X_e and n shows that continuity of expansion does not and should not exist except under very special conditions.

141 *Division of Feed for Applying Hirn's Analysis.* One of the requirements of Hirn's analysis is that we know exactly how much steam was admitted to each end of a cylinder. These amounts are not usually equal in practice, so an assumption must be made to cover the needs of the case.

142 The usual assumption is to divide the feed between each end of a cylinder in the ratio of the values of the mean effective pressure shown by the PV-diagrams from the two ends. This assumption is probably not far from the actual division in most cases, and is the best that can be done under the circumstances.

143 In the light of the facts presented in this investigation, this feed may now be divided on a more rational basis. It has been found that the presence of the piston rod in only one end of the cylinder has no appreciable effect upon the value of n. This fact enables us to divide the feed according to the volumes filled and the values of X_e as determined by the resulting values of n. This method is believed to be the closest solution obtainable in the case where the supply for each end of the cylinder cannot be separately measured.

144 *Computing the Weight of Steam Retained in Compression.* A carefully made investigation indicates,[1] when saturated steam is used, that the steam in the cylinder is dry or even very slightly superheated at the closure of the exhaust valve. As shown by the logarithmic diagrams, leakage of the steam in compression continues until the exhaust valve, in closing, has acquired considerable seal.

145 The point which is selected to compute the volume of steam retained in compression generally lies between the points A to B and C to D of Fig. 9. As the weight of steam retained is yet decreasing,

[1] George Duchesne, Revue de mécanique, July 1899: quoted in Power, Jan. 10, 1911, p. 71.

the point selected nearly always accounts for more steam than was actually retained. In other words, we do not find the value of X_e to be as high as it actually is.

146 The following method shown in Fig. 9 has been adopted in the present investigation. The straight line of the compression curve on the logarithmic diagram, or the line of constant weight of steam mixture, is prolonged dotted as shown to the back pressure. The intersection of this prolonged line with the back pressure line extended is taken as the volume of dry steam retained in compression. This method almost always gives less steam retained in compression than the ordinary method, and is believed to be rational in the present state of knowledge of this subject.

CONCLUSIONS

147 The following conclusions have been drawn from the results of the investigation relating to all elastic media, as applicable to the cylinders of reciprocating engines using elastic media and having, as a part of the working cycle, an expansion, a compression, or both:

a The indicator diagram, taken by means of a correct reducing motion and with a reliable indicator, contains the evidence necessary for a very complete and useful analysis of cylinder performance.

b The logarithmic diagram, derived from the indicator diagram, discloses a new and very complete analysis of cylinder performance.

c Free from the influences of leakage, wrong clearance, wrong location of the line of zero pressure, and excessively low speed (principally in steam engines), expansion or compression of an elastic medium takes place substantially according to the law, $PV^n = C$.

d The clearance of cylinders may be found by graphical trial in the logarithmic diagram to within 5 per cent to 10 per cent of the clearance volume, depending as the clearance itself varies from 20 per cent to 2 per cent, respectively, of the piston displacement.

e The cyclic events, even though entirely obscure in the indicator diagram, may be located on the logarithmic diagram (when plotted on logarithmic paper of 5 in. per square) to within $\frac{1}{16}$ in., this quantity being the equivalent of about $\frac{1}{32}$ in. when retransferred to the indicator diagram.

f Leakage (if appreciable) may be reliably detected from the logarithmic diagram, and may, in some cases, be approximated in volume.

g The weight of steam retained in compression should be obtained from the logarithmic diagram by prolonging the line of constant weight of steam mixture to the back pressure line extended; the intersection of these two extended lines is the volume of steam which is retained.

148 Acknowledgment is made to Prof. G. A. Goodenough, through whose kind offices permission was secured to undertake the investigation, and who at all times has given his hearty encouragement to the work; also, to Dean W. F. M. Goss, Prof. O. A. Leutwiler, C. M. Garland, Purdue University, and to the various firms who have furnished indicator diagrams and tests for analysis.

APPENDIX

PART 1 METHOD OF CONSTRUCTING THE LOGARITHMIC DIAGRAM

149 The logarithmic cross-section paper used in this investigation consists of four squares arranged two each way. These squares are 5 in. each way, making the four squares together 10 in. each way. The use of four squares enables values to be plotted ranging from 0.1 to 10.0, 1.0 to 100.0, etc., thus giving a range of ten times the values obtainable if only one square were used.

150 *Construction of the Logarithmic Diagram.* The coördinates of the *PV*-diagram are proportional to pressure and stroke, the latter being proportional to the volume displaced by the piston. The coördinates of several points on the *PV*-diagram are found in terms of absolute pressure, preferably in pounds per square inch, and absolute volume, preferably in cubic feet, these being the units used in steam tables.

151 The method of transferring the *PV*-diagram to the logarithmic form is described in detail for the diagrams of test 30, given in Fig. 15. The method of drawing the pressure ordinates is shown in Fig. 15, Crank End. The diagram is shown in outline by *ABYX*. Perpendiculars *QR* and *EX* are drawn to the atmospheric line *EQ*, and pass through the extreme stroke positions of the diagram. The distance *EQ* is then the length of the diagram. *OM* is laid off perpendicular to the atmospheric line *EQ* (extended) which was drawn by the indicator pencil. *OM* is the line of zero volume, and is drawn at a distance *FE* from the admission end *EX* of the diagram, the distance *FE* being the same length in per cent of the line *EQ*, or length of the diagram, as the proportion that the per cent of clearance, or waste space, of the cylinder bears to the piston displacement. In this case the length of the diagram is 3.99 in., and the clearance is 7.04 per cent. The length *FE* is therefore 0.0704 multiplied by 3.99, or 0.281 in. *ON* is the line of zero pressure and is drawn a distance *FO* below the atmospheric line, to the scale of the spring used in obtaining the *PV*-diagram. This distance is proportional to the barometer reading, corrected for temperature, prevailing at the day and place of the engine test. In this case the corrected barometer reading was 14.2 lb. per sq. in. absolute, so the distance *FO* is $\frac{14.2}{79.0}$ or 0.180 in.

152 From *ON* points are laid off on *QR* and *EX* corresponding to the absolute pressures at the intervals where it is desired to read off the corresponding volumes. Fine lines are drawn connecting similar pressure points, as 19.8–19.8, 29.1–29.1, etc. The volumes *G–A*, *G–B*, *H–D*, *H–C*, etc. are read off in hundredths of an inch to the nearest half hundredth. The tabular form used in this investigation is given in Table 6 for the diagrams of Fig. 15 taken in test 30. Thus the length *G–A* is read off as 0.66 in., and is given under the column for 19.8 lb. pressure, headed Compression, for the crank and diagram. The volumes in inches are then multiplied by the constant ratio which 1 in. of length of the diagram bears to the displacement of the piston. From Table 6 it is seen that the piston displacement of the crank end is 1.523 cu. ft., and the length of diagram 3.99 in., hence, the ratio is $\frac{1.523}{3.99}$ or 0.382 cu. ft. of piston displacement per inch

of diagram length. The length G–A in cu. ft. of displacement now becomes 0.66 multiplied by 0.382, or 0.252 cu. ft., the volume of steam present at this point. This process is repeated at intervals until the coördinates of from 10 to 30 points are determined. In the diagram shown in Fig. 15, the coördinates of 18 points were found in each diagram.

153 The coördinates of P and V are then plotted on logarithmic cross-

TABLE 6 CONSTRUCTION OF THE LOGARITHMIC DIAGRAMS OF TEST 30 †

Number	Head End					Crank End				
	Absolute Pressures, Lb. per Sq. In.	Volumes				Absolute Pressures, Lb. per Sq. In.	Volumes			
		In.		Cu. Ft.			In.		Cu. Ft.	
		Compression* X to A	Expansion* Y to B	Compression	Expansion		Compression	Expansion	Compression	Expansion
1	133.0	0.39	1.18	0.156	0.470	134.3	0.30	1.205	0.115	0.4605
2	113.4	0.31	1.36	0.124	0.5425	115.3	0.28	1.39	0.107	0.531
3	95.0	0.31	1.625	0.124	0.649	96.3	0.28	1.65	0.107	0.6305
4	76.3	0.31	2.02	0.124	0.8055	77.3	0.28	2.06	0.107	0.7875
5	57.1	0.31	2.705	0.124	1.080	58.1	0.295	2.76	0.113	1.055
6	38.9	0.375	4.03	0.150	1.610	38.7	0.385	4.12	0.147	1.572
7	29.4	0.48	4.22	0.192	1.685	29.1	0.485	4.26	0.185	1.626
8	19.5	0.70	4.26	0.279	1.699	19.8	0.66	4.20	0.252	1.600
9	15.0	1.10	3.06	0.439	1.220	15.0	0.93	3.77	0.355	1.440

* Letters refer to Fig. 15, Crank End.
† Final results given in Table 1.

	Head End	Crank End
Length of indicator diagram, in.	3.95	3.99
Ratio of clearance to piston displacement, same end	0.0789	0.0704
Length on diagram proportional to clearance ratio, in	0.312	0.281
Length of diagram, plus clearance, in.	4.26	4.27
Piston displacement (cylinder 12.02 in. by 24 in.), cu. ft.	1.575	1.523
Clearance volume, cu. ft.	0.124	0.107
Displacement plus clearance, total volume, cu. ft.	1.699	1.630
Ratio, cu. ft. per in. of length on diagram, cu. ft.	0.399	0.382
Scale of indicator spring per in. of ordinate, lb.	77.0	79.0

section paper, as shown in Fig. 9, which are the logarithmic diagrams derived from the PV–diagrams of Fig. 15. The points plotted in Fig. 15 are taken from the columns headed Cu. Ft. at the pressures shown. A smooth curve is drawn through the points thus plotted, and the diagram is in shape to be studied.

154 *Examples of Logarithmic Diagrams from Various Types of Engines.* The logarithmic diagrams obtained by plotting the indicator diagrams as already described, are of a distinctly different form from either the PV–diagram or the temperature-entropy diagram. While the various typical forms of PV–diagrams assume rather different forms after plotting, yet the resultant figures retain in a general way the peculiar characteristics of each PV–type except that these peculiarities are exaggerated.

155 Various typical PV–diagrams are given in Figs. 16–25, which include examples from many types of engines using steam, gas, air and ammonia as the

active media. The values of n for the curves are given in each figure and it will be noted from these diagrams how closely the law $PV^n = C$ holds for the expansion and compression curves from a great variety of engines using different media.

156 It has long been known that adiabatic expansion, or compression, of any elastic medium takes place substantially according to this law with the different values of n for different media. In practice, however, expansion is never adiabatic, but is changed in character by the presence of the metal surrounding the working medium, and by imperfection of mechanism. The question then arises as to how this expansion has been changed in character and whether it still obeys the law $PV^n = C$.

157 Many investigators have examined the curves from actual diagrams to clear up this point for steam.[1] These include those who find that expansion in steam cylinders takes place substantially according to the law $PV^n = C$, but that n varies in value between wide limits in different cases.

158 An examination was made of the curves of 296 diagrams from the cylinders of 47 different engines using steam, gas, air and ammonia to investigate this point. As a result, it may be stated that, in the cases of the great majority of engines using elastic media, expansion and compression take place substantially according to the law $PV^n = C$. Certain exceptions, however, have been found, and the cases studied.

159 *Cases where the Law $PV^n = C$ does not hold.* The curves of expansion and compression, obtained from PV–diagrams, do not always follow the law $PV^n = C$. This fact is due to several causes, some of which have been definitely determined:

160 (*a*) *Wrong Clearance, or Wrong Location of the Zero Line of Pressure.* The law, $PV^n = C$ is only true where P and V are measured in absolute units. The clearance must be accurately determined in order to give absolute values of V. The scale of the spring, for the PV–diagram analyzed must be known, and also the atmospheric line drawn by the indicator, in order to locate the zero line of pressure. The units used for P or V may be of any denomination, but they must be measured from the zero of P and V.

161 When a curve $PV^n = C$ is plotted on logarithmic paper, the resultant curve is a straight line. The value of n as already explained, is the slope of this line measured from any two points. When the values of P and V are not absolute values, this curve is no longer a straight line, but becomes a curve of the second degree. The value of n being the slope obtained from two points on this curve, is no longer constant for all parts of the curve, but varies from point to point.

162 Therefore, when PV-diagrams are transformed to logarithmic diagrams, the values of both P and V must be measured in absolute units. When these values are not in absolute units, the resulting curve is not of the form $PV^n = C$, and therefore is not a straight line on logarithmic paper. The form of the curve obtained, when the values of V alone are not in absolute units, is given in Fig. 10-b.

[1]Technical Thermodynamics, Zeuner, vol. 2, p. 111.

Recherches experimentales, Leloutre, Bulletin de la Société industrielle du Nord de la France 1874.

Zur Theorie des Indikatordiagrammes, Luders, Zivilingenieur, 1881, vol. 27, p. 225.

The Steam Engine, Perry, p. 106.

163 (*b*) *Leakage.* The law $PV^n = C$ is applicable only in cases where the weight of the working medium remains practically constant during any expansion or compression. When this weight changes materially either by leakage into, or out from, the cylinder containing the medium the resulting curve no longer obeys the law, and it becomes a curve in the logarithmic diagram. This fact is very clearly shown in the curves of the logarithmic diagram derived from cylinders in which large leaks were known to exist. The examples showing this condition have already been discussed.

164 (*c*) *Low Speed in Steam Engines.* Very low speeds of rotation, together with very low piston speeds, will cause the compression curve to deviate from the law $PV^n = C$. The most common cases of this effect are seen in the "hook" or excessive condensation, near the upper end of compression curves. These hooks are found almost altogether in small engines having very low piston speeds, and in larger engines with small clearance, having very low rotational speeds, as in pumping engines.

165 Diagrams containing hooks in the compression curves have been published by Professor Dwelshauvers-Dery[1] from the experimental engine at Liege. This is an example of a very low speed in a small engine. The size was 12 in. by 24 in. and the speed from 30 to 60 r.p.m. That this hook was caused by low speed and consequent excessive condensation, was proved in a measure in the case of the engine used by the author. This engine was also 12 in. by 24 in. but was operated at from 90 to 150 r.p.m. In no case was a hook in the compression curve obtained during the tests, although some 1600 diagrams were taken. A set of diagrams from these tests is shown in Fig. 15, and shows no sign of a hook. On one occasion, however, the diagram shown in Fig. 26 was obtained. This was taken just after the engine was started from cold, and had been brought up to a speed of 120 r.p.m. In conjunction with the other diagrams obtained from this engine in regular operation, this hook is believed to be due to excessive condensation while the cylinder was comparatively cold.

166 The three causes just treated are believed to be the important conditions that cause the curves in practice to depart materially from the law $PV^n = C$. Sometimes only one of these conditions is present, while in other cases, a combination of these may influence the resulting curves. The separation of these conditions by their effect on the curves has already been treated.

PART 2 PRINCIPAL DIMENSIONS OF ENGINE USED IN TESTS

a Type—Horizontal single-cylinder, double-eccentric, non-condensing, variable speed, heavy-duty frame, Reynolds Corliss engine

b Class—Belt drive for mill work

c Maker—Allis-Chalmers Company, Milwaukee, Wis.

d Rated Power of Engine—100 h.p. at 115 lb. initial pressure above atmosphere on indicator diagram, $\frac{1}{4}$ cut-off, and 120 r.p.m.

e Cylinder dimensions

a Bore (measured while hot), in	12.02
b Stroke, in	24.00
c Diameter of piston rod, in	$2\frac{3}{16}$

[1]Power, June 28, 1910, p. 1165.

f Clearance—In per cent of volume displaced by piston per stroke
  a Head end 7.89
  b Crank end 7.04
g Speed—Controlled by fly-ball governor with variable gear ratio between main shaft and governor, giving any engine speed from 20 to 160 r. p. m. Ususal speed 120 r.p.m.

167 The cylinder of the engine is not steam-jacketed on the ends but is partly jacketed on the barrel by the steam chest, the latter covering about one-sixth of the barrel surface. The exhaust passages are separated from the lower part of the cylinder barrel by a dead air space formed in the cylinder casting.

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER RESERVOIRS DUE TO COOLING EFFECTS OF AIR AND EVAPORATION

By W. B. Ruggles

ABSTRACT OF PAPER

The tests described in this paper were made to determine the heat radiation from a cooling reservoir of about 6½ acres, at the Crescent Portland Cement Company's plant at Wampum, Pa. The amount of heat delivered to the reservoir from the engines and compressors was measured by hourly readings for power and vacuum and the reduction of temperature, due to pumping in fresh water from the Beaver River, and by rain was also taken. By frequent readings of temperature of tail water, intake water and air, the cooling effect of the air on a unit of surface of water per unit of difference of temperature was easily determined.

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER RESERVOIRS DUE TO COOLING EFFECT OF AIR AND EVAPORATION

By W. B. Ruggles, New York
Member of the Society

The tests described in this paper were made with the idea of determining if possible, a fairly reliable factor by the use of which the necessary size of a cooling reservoir for condenser water could be predetermined for any assumed power and weather conditions.

2 In 1908, while designing the mill for the Crescent Portland Cement Company at Wampum, Pa., it was found necessary to install a cooling device for the condensing water from the engines. The mill stands on a level plateau about 110 ft. above the Beaver River and to pump water from the river for condensers meant a considerable expense for power. On the side of the plateau furthest from the river was a natural depression alongside the Erie division of the Pennsylvania Railroad which offered a desirable location for a reservoir (Fig. 1). After a diligent search, no data were found as to how large a reservoir had to be to give the necessary cooling effect to the water, so that it might be used continuously. A dam, however, was built 275 ft. long and 18 ft. high, of skeleton reinforced concrete construction which would impound about $6\frac{1}{2}$ acres of water (Fig. 2).

3 Three tests were made during the year 1911, each of one week's duration in order to determine the heat radiation from the surface of the reservoir. These tests were made in May when the temperature was moderate, in July when the temperature was high and in November when the temperature was low. Readings were taken of the temperature of the river water, the intake water to the power house, the tail water from the condenser and of the air. The vacuum,

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

the power, the amount of water pumped from the river to the reservoir, and the rainfall were also recorded. These data were taken by James Toler, Chief Engineer of the Crescent plant and by the writer's assistant, John E. Mason. Unfortunately the temperature of the rain was not taken, but it has been assumed the same as the temperature of the air, so that the figures for that are only approxi-

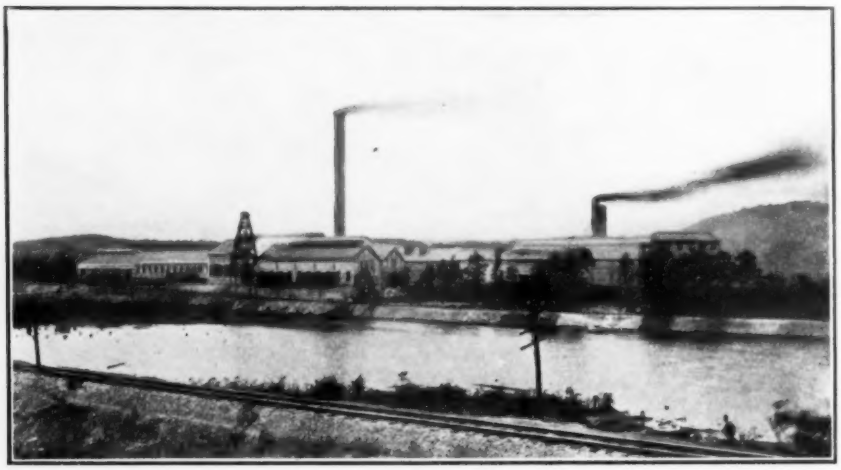

FIG 1 VIEW OF THE CRESCENT PORTLAND CEMENT COMPANY PLANT SHOWING RESERVOIR

mate, but a difference of 10 deg. in the temperature of the rain would not alter the result by more than $\frac{2}{10}$ per cent.

4 The engines used in the power house are three Ball and Wood compound condensing engines, 26 in. by 52 in. by 36 in. The air compressors are two compound Laidlaw-Dunn-Gordon compressors, the steam cylinders of which are 12 in. and 21 in. in diameter respectively, the air cylinders $11\frac{1}{2}$ in. and 24 in. respectively, the stroke 24 in. The steam from all of these is condensed in an Alberger barometric condenser equipped with a dry vacuum pump and the circulating water supplied by a direct-connected centrifugal pump driven by a synchronous motor; an engine-driven circulating pump and a second dry vacuum pump being installed as reserve.

5 The water from the condenser flows about 50 ft. through a tile pipe into the east side of the reservoir about 100 ft. from the north end. A dike was built south of this inlet to the reservoir extending about 50 ft. toward the center of the reservoir and then

north nearly to the north end. This compels the circulating water to flow up and around the end of this dike and down toward the dam, a distance of about 1100 ft., before it reaches the intake to the power house.

6 Water to supply the reservoir is pumped from the river by a centrifugal pump direct connected to a 150-h.p. motor. The pump delivers an average of 2324.25 gal. of water per minute and it is found that about 8 hours pumping per week is sufficient to make up for the evaporation and seepage in the reservoir. The water from the river pump is delivered into an open well situated on top of the plateau where the mill is located. It flows by gravity from

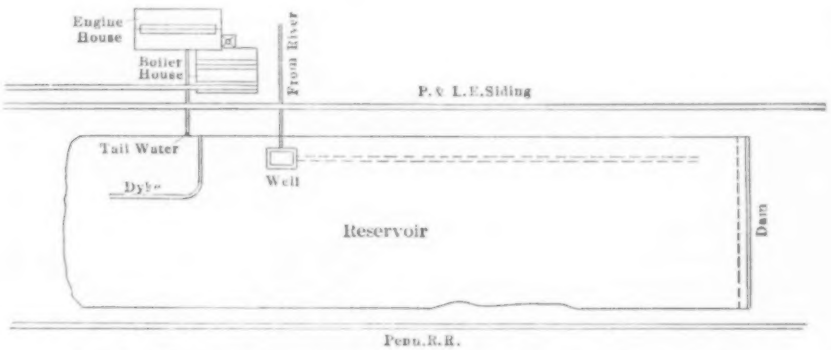

FIG. 2 DIAGRAM SHOWING 6½-ACRE RESERVOIR AND CONNECTIONS

this well through a 24-in. tile pipe into another concrete well located in the reservoir about 800 ft. north of the dam. From this well it flows through a No. 24 tile pipe laid on the bottom of the reservoir to within 6 ft. of the dam. The intake water for the boiler feed and condenser is drawn from this well so that the water must flow up through the concrete tile pipe from the bottom of the reservoir at its deepest point. While water is being pumped from the river, there is a flow south through this tile pipe and when not pumping, the water is drawn in the opposite direction through it.

7 The average horsepower developed in the engines during the 21 days of these tests was 2446. Incidentally, it may be stated that there was an average of 3062 bbl. of cement manufactured per day during the same test, or 1 bbl. of cement for 0.8 h.p.

8 The results of the tests during these three weeks are as uniform as could be expected; the heat lost per square foot of surface being slightly less per 1 deg. difference of temperature in cold weather than

in warm weather. This difference is undoubtedly due to an increased amount of evaporation during the warmer period.

9 The average humidity for the different weeks during which the tests were made was 58.5 per cent in May, 62.3 per cent in July and 71.2 per cent in November.

10 Table 1 gives the results in detail of the tests made.

TABLE 1 TEST OF HEAT RADIATION FROM SURFACE OF CONDENSER WATER RESERVOIR AT PLANT OF CRESCENT PORTLAND CEMENT CO., WAMPUM, PA.

Area of reservoir, 288,000 sq. ft.; average depth of reservoir, 5.36 ft.; capacity of reservoir, 1,543,680 cu. ft. = 96,480,000 lb.

Date of Tests	Week ending May 7, 1911.	Week ending July 12, 1911	Week ending November 27, 1911.
Amount of water pumped from river, lb.	10458956	29050875	4648140
Average temperature of river water, deg. fahr	57.5	77	36
Average temperature of intake to power house, deg. fahr	72.75	91.43	61.71
Average temperature of tail water from condenser, deg. fahr	101.36	129.43	90.71
Average temperature of reservoir, deg. fahr	87.05	110.00	76.71
Average temperature of air, deg. fahr.	51.00	78.43	33.30
Average difference of temperature between water and air, deg. fahr	36.05	31.57	43.41
Change in temperature of reservoir during test, deg. fahr	0.25	7.00	2.00
Three 26 in. by 52 in. by 36 in. compound condensing engines, h. p-hr.	397807	411320	426936
Two 12 in. by 21 in., 11½ in. by 24 in. by 24 in. air compressors, h.p-hr	46536	46536	46536
Average vacuum at 30 in barometer, in.	26.4	22.2	26.6
Average water consumed by engines, per h.p. per hr., lb	14.46	15.64	14.41
Average water consumed by compressors per h.p. per hr., lb	18.85	20.12	18.83
Steam condensed by engines, lb.	5752289	6433045	6145148
Steam condensed by compressors, lb.	877204	936314	876273
Latent heat of steam condensed, lb.	1024.7	1007.1	1026.0
Heat delivered to reservoir by engines, b.t.u	5894370000	6478720000	6304922000
Heat delivered to reservoir by compressors, b.t.u	898871000	942961000	899056000
Heat to raise river water to average temperature of reservoir, b.t.u	309062000	958679000	189226000

TABLE 1—*Continued*

Date of Tests	Week ending May 7, 1911	Week ending July 12, 1911	Week ending November 27, 1911
Heat given up or retained in reservoir during test, b.t.u.	24120000	675360000	192960000
Heat reduction in reservoir due to rain, b.t.u	21630000	56700000	46200000
Heat absorbed by air and evaporation during seven days, b.t.u.	6438429000	5730942000	6564509000
Heat absorbed by air and evaporation. per sq. ft. of surface seven days, b.t.u.	22356	19899	23495
Heat absorbed by air and evaporation per sq. ft. per hr., b.t.u	133.1	118.4	139.8
Heat absorbed by air and evaporation per sq. ft. per hr., per 1 deg. difference b.t.u	3.69	3.71	3.22

11 It appears from the tests that under ordinary conditions, in the northern part of the United States with engines using 15 lb. of water per h.p-hr. and a vacuum of 26 in., a reservoir having a surface of 120 sq. ft. per h.p. would be ample for cooling and condensing water.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

LES NOUVELLES EXPÉRIENCES DE M. EIFFEL AU LABORATOIRE DU CHAMP-DE-MARS. *L'Aérophile*, February 15, 1912. 5 pp., 13 figs. *e*. An account of various *experiments of Eiffel on air resistance*. The article is an abstract of Eiffel's book, La résistance de l'air et l'aviation, published in Paris last year.

Air Machinery

TOURBO-SOUFFLANTE DE GRANDE PUISSANCE POUR ACIÉRIE. *Le Génie Civil*, February 10, 1912. 1/2 p. *d*. Description of a *turbo-blower* installed by the Brown, Boveri Company of Baden, Switzerland, at the plant of the Metallurgical Company of Sambre and Moselle, at Montignies-sur-Sambre, France, to supply the draft for four 15-ton Bessemer converters. The blower is able to compress from 150 to 800 cu. m. (5300 to 28,250 cu. ft.) per minute to pressures from 0.4 kg/qcm (5.6 lb. per sq. in.) to 2.5 kg/qcm (35.5 lb. per sq. in.). The blower is driven by a turbine of 3750 maximum h.p., and has the maximum speed of 2600 r.p.m. In comparison with reciprocating blowers, the turbo-blower is remarkably compact: the foundations occupy only 38 sq. m. (406 sq. ft.), while a reciprocating blower would occupy about 168 sq. m. The expenses of upkeep and lubrication are said to be very low, while the regularity of operation materially increases the efficiency of the converters, and makes unnecessary the use of air-pressure regulators in the tuyeres. When no blowing is required, the blower rotates at about 800 r.p.m.; when the signal for blowing is given, the pressure is raised to 1.9 kg/qcm (27 lb. per sq. in.) in about 10 to 15 seconds; at the end of the blowing the pressure is raised to 2 or 2.5 kg/qcm (28.4 or 35.5 lb.

per sq. in.). The pressure and quantity of air delivered is regulated with perfect certainty by changing the speed of the blower, no auxiliary pumping being required at the beginning.

Brakes

NEUE BREMSEN, INSBESONDERE FÜR GIESSEREIKRANE, Wintermeyer. *Giesserei-Zeitung*, February 1, 1912. 2½ pp., 3 figs. *d.* Description of new *brakes for foundry cranes*, both for single-phase asynchrone and for two-phase motors.

Fuel

EINE NEUE BRIKETTPRESSE. *Braunkohle*, February 9, 1912. 3½ pp., 6 figs. *d.* Describes a new type of briquette press built by Ehrhardt and Schmer, in Saarbrücken, Germany. The crankshaft is supported by bearings belonging to the body of the press, and carries on one of its ends the fly-wheel, and on the other the disc-crank and crank-pins of the steam engine, the press-rod being connected to it in the middle. The press-rod has a forced-feed lubrication, with the surface pressure so slight that no water cooling is necessary. The valve gear of the steam engine is arranged as follows: the admission valves are at the top of the steam cylinder, actuated by a wedge-gear, its reciprocating motion being transmitted to it from an eccentric on the engine-shaft by a rocking lever and rod. The valve gear is positive, and the valves open and close rapidly, but with a somewhat softer motion in closing. The exhaust is placed on a side at the bottom of the steam cylinder, so as to let out absolutely all the water of condensation, and is actuated, independently of the admission valve, by a piston valve of special construction. The advantages of this press are: there is only one fly-wheel; the bearings for the crankshaft and the main pressure rod are easily accessible; no spur or bevel wheels, and many less bolts are used, the gear transmission being throughout by means of eccentrics. One of the illustrations shows how a press of the older type is changed to this construction.

VERWERTUNG DER MOORE DURCH GEWINNUNG VON KRAFTGAS UND AMMONIAK, A. Frank. *Journal für Gasbeleuchtung*, January 20, 1912. 4 pp. *p.* An account of the work of the Hannoversche Kolonisations- und Moorverwertungsgesellschaft, which bought about 1000 ha (2471 acres) near Osnabrück, Prussia, and are producing *electric current by producer-gas engines* from power-gas obtained from Mond generators, *with peat as fuel*. For 1000 h.p. per year, produced at the rate of 8000 h.p. per hour, the consumption of peat is taken to be about 4 ha (9.88 acres), the layer of peat being 3 m. (9.8 ft.), and a ton of dry peat producing at least 650 to 700 h.p-hr. The soil from under the peat is treated in the same manner as in Holland and converted into high grade agricultural land. Power gas can be obtained from peat containing as much as 50 to 55 per cent of water, the gas having about 1400 WE (160 b.t.u. per cu. ft.). With higher content of water, both the content of carbon dioxide and volume of gas rose, but heat content decreased. An amount equal to 2.4 cbm (84.7 cu. ft.) per effective horsepower of the normal gas (i. e. obtained from peat with

water content not above 55 per cent) was required. As a by-product, about 40 kg. (88 lb.) of ammonium sulphate per ton of dry peat was obtained.

Internal Combustion Engines

IL PETROLIO NELLA NAVIGAZIONE. *L'Elettricista*, February 15, 1912. 1¼ pp. *g.* Discussion of the *use of oil fuel in propulsion of vessels* and description of the vessel Vulcan propelled by Diesel engines built for the Danube trade for the Astra Romana Co. The article is based on one by L. Erbiceanu in the *Monitore del petrolio rumeno*, August 2, 1911. The tank-ship Vulcan for the transport of oil was built in Amsterdam, Holland, and equipped with a 6-cylinder, 4-cycle, 500 h.p., Diesel engine, with a vertical compressor delivering air at 60 kg/cm (853 lb. per sq. in.) pressure. The motor can be reversed by a hand lever, the engine being previously stopped, and the whole time necessary for this operation does not exceed 20 seconds. The consumption of oil (Roumanian) was found to be about 0.2 kg (0.44 lb.) per h.p-hr., at a normal speed of 8 miles per hour. The operation of the motor proved to be satisfactory, noise not excessive, but vibration more noticeable than in the case of a steam engine.

SEEFISCHEREI-MOTOREN, F. Romberg. *Dinglers Polytechnisches Journal*, February 3, 1912. (Cp. The Journal, March 1912, p. 411). *d.* End of a series of articles on the *design of motors for fishing craft*. For a detailed abstract see *Engineering*, March 1, 1912.

WIEDER EINE EXPLOSIONSTURBINE. *Der praktische Maschinen-Konstrukteur*, February 1, 1912. 1 p., 1 fig. *d.* Description of a *gas turbine* lately built by F. J. E. Johansson in Stockholm, Sweden. This turbine has a number of explosion chambers placed side by side, each provided with two admission and one exhaust valves which are so arranged that when an exhaust valve opens in one chamber, it throws open the admission valve in the next, and permits it to be filled with an explosive mixture. For a complete translation of this article see *The Automobile*, March 7, 1912.

DER BRÜNN-KÖNIGSFELDER GLÜHKOPF-ZWEITAKTMOTOR. Ernst Neuberg. *Die Gasmotorentechnik*, February 1912. 7 pp., 12 figs. and 1 plate of drawing. *d.* Description of the *Brünn-Königsfelder two-cycle motor*. The author divides all internal-combustion engines into two classes: those above 15 h.p., and those below. The man who buys a small engine, say of 5 h.p., will probably in about two years, as his business grows, need an 8-h.p. one, and in about two more years a 12-h.p. He will therefore from time to time come to the place where he bought the engine, and want to have it exchanged for a larger one. The man who buys a large engine, say of 100 h.p., when he wants more power, will as a rule not exchange his engine for a larger one, but buy another. Wtih the buyer of the small engine the intial cost is therefore a primary consideration, while in selecting a large engine more attention is paid to cost of operation and yearly amortization required. In the first case therefore small engines have to be as cheap as possible, while in the second a good deal more attention is paid to workmanship. The following are the principal characteristics of the two classes:

FIRST CLASS.	SECOND CLASS
Lubrication	
Forced lubrication for the crank bearings, crank pin, piston pin, and cylinder, wick oiling being used only for thrust bearings in motor boats.	Forced lubrication for the cylinder and piston pin. Wick oiling for crank and thrust bearings in motor boats. Bearings with oil rings for the main bearing of crankshaft in stationary motors.
Governing	
Centrifugal governor regulating the supply of combustible in proportion to power required.	Same as in First Class.
Starting	
The flywheel is rotated through half a revolution by hand.	Without compressor by air pressure of 8 to 10 atmospheres, the compressed air being replaced from the working cylinder.

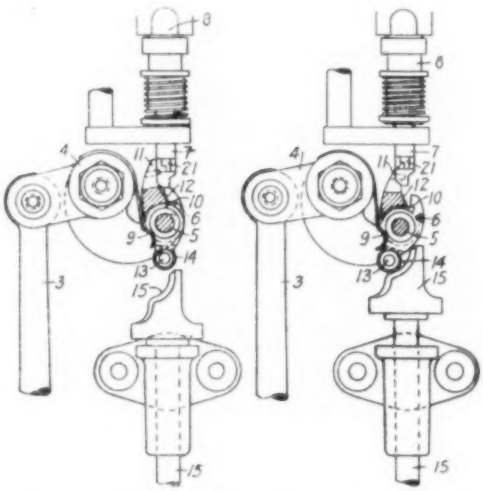
The article proceeds to describe in detail the system employed by the factory to standardize the production as far as possible. This system is based on the same principle as that described in The Journal, January 1912, p. 132. The details of specifications of various types of engines are also described.

The reversing of the motor-boat type of these engines is arranged as follows: The fuel pump is under the influence of two governors: the centrifugal governor regulating the speed of the engine, and a second, inertia, governor which begins to act at reversing, and only when the speed has gone down considerably. The action of the reversing lever produces therefore the following effects: (*a*) it cuts out the centrifugal governor and the fuel pump; (*b*) it cuts in the inertia governor which forces the pump to deliver to the engine two or three times as much fuel as before, the speed being considerably decreased, and at an earlier moment than usual; this leads to a premature explosion which reverses the direction of rotation; (*c*) it then cuts out the inertia governor, and cuts in the centrifugal governor and fuel pump.

Figs. 1 and 2 show in detail the contruction of the single-cylinder motor. On the motor-shaft is placed a centrifugal governor of the usual construction which acts on the eccentric whenever the speed or output of the motor is changed. This eccentric is connected by the rod 3 with lever 4 in such a way that the amplitude of the lever changes with the change in the travel of the eccentric. The catch 6 on pin 5 of lever 4 transmits the motion of the lever to the plunger 7 of the fuel pump 8, so that an increase in the travel of the eccentric causes an increase in the amplitude of the lever, and that leads to a greater swing of the plunger in the fuel pump, and larger supply of fuel. The spring 9 and stop-pin 10 serve to keep the catch 6 in place. On the pin 5 is also the catch 11 of the inertia governor, the projections 12 preventing the catch 11 from

moving to the right and thus engaging with the plunger 7. These figures show the position of the parts for the normal action of the motor.

PERFECTIONNEMENT DES MOTEURS À COMBUSTION INTERNE PAR LE RECHAUFFAGE PRÉALABLE DE L'AIR, A. Nougier. *Le Génie Civil*, February 24, 1912. 3 pp., 1 fig. *dtA*. Beginning of a series of interesting articles on the *use of preheated air in internal-combustion engines* of the Diesel type. The incentive to seek for an engine which would economically burn heavy oils was due to the heavy duty on petroleum in France, making the use of ordinary Diesel engines very uneconomical. Carels Brothers in Gand, Belgium, tried to get around this by feeding their engines with tar-oil, but injecting some petroleum before the end of the compression stroke, thus using the petroleum as a sort of ignition for the tar-oil. This necessitated, however, the installation of a double valve



FIGS. 1 AND 2 BRÜNN-KÖNIGSFELDER TWO-CYCLE MOTOR

gear, and so complicated the working of the engine that it had to be given up. Aside from the comparative cost of petroleum and heavy oils a system that would permit of doing away, even partly, with the high compressions used in the Diesel engine would be of great advantage by dispensing with the air compressor which increases the first cost of the engine and takes as much as 7 per cent of the power delivered to the engine shaft, and by making less important than at present the great strength of the cylinders and elaborate arrangements for keeping them and the valves thoroughly air tight. The author claims that this can be done by *preheating the air drawn into the engine*. He bases his claim on the principle that *a slight increase in temperature of the air at the beginning of the compression stroke results in a large increase of temperature at its end*, and proves it in the following way: Since pure air is practically a perfect gas, and the compression practically adiabatic, either of the following three equations must hold:

$$\frac{p_1 v_1}{T_1} = \frac{p_o v_o}{T_o} \quad \ldots\ldots [1]$$

$$p_1 v_1^{1.41} = p_o v_o^{1.41} \quad \ldots\ldots [2]$$

$$p_1^{0.71} v_1 = p_o^{0.71} v_o \quad \ldots\ldots [3]$$

By dividing, member by member, [1] by [2], the relation

$$T_1 = T_o \left(\frac{v_o}{v_1}\right)^{0.41} \quad \ldots\ldots [4]$$

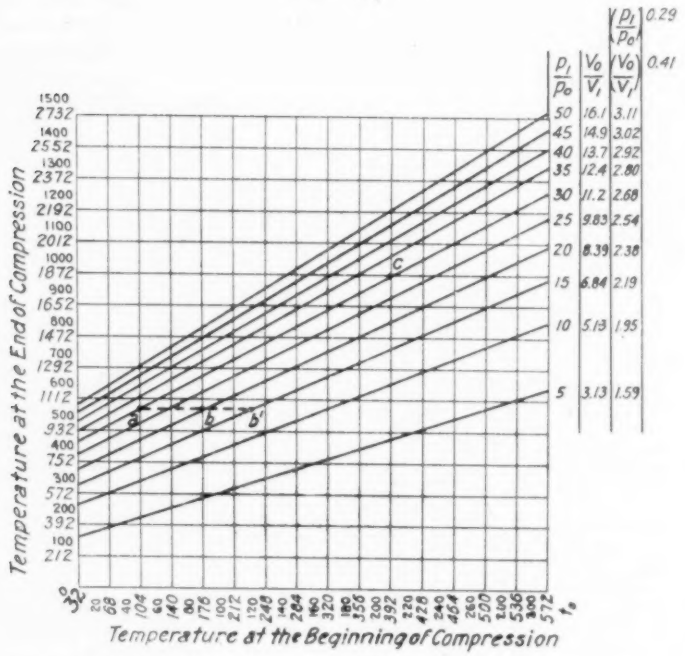


FIG. 3 ADIABATIC COMPRESSION—RISE OF TEMPERATURE CURVES FOR AIR
The small, vertical figures indicate deg. cent.; the larger, inclined figures, deg. fahr.

is obtained which gives the final absolute temperature T_1 of air taken at initial absolute temperature T_o and subjected to adiabatic reduction of volume $\frac{v_o}{v_1}$.

By dividing, member by member, [1] by [3], the relation

$$T_1 = T_o \left(\frac{p_1}{p_o}\right)^{0.29} \quad \ldots\ldots [5]$$

is obtained which gives the final absolute temperature T_1 of air taken at initial absolute temperature T_o and subjected to adiabatic compression $\frac{p_1}{p_o}$. Fig. 3 gives a set of curves expressing the relations [4] and [5] for certain values of $\frac{v_o}{v_1}$ and $\frac{p_1}{p_o}$. For each given value of $\frac{v_o}{v^1}$ or $\frac{p_1}{p_o}$, equations [4] and [5] represent straight lines, and all these straight lines meet at a point having the coördinates -273 and

-273 (deg. cent.). The third row of figures on the upper right corner indicates the values of $\left(\frac{p_1}{p_o}\right)^{0.29}$ or $\left(\frac{v_o}{v_1}\right)^{0.41}$ which would be equal to the angular coefficient of the line if the scale of the abscissæ were equal to that of the ordinates. This graph facilitates the obtaining of the final temperature t when the initial temperature to and adiabatic compression is known and can be expressed either in the form of $\frac{p_1}{p_o}$ or as $\frac{v_o}{v_1}$. Thus, if the initial temperature of the air was 40 deg. cent. and it was compressed to 30 atmospheres, its final temperature will be 566 deg. cent.; but the same temperature may be attained by compressing the air to only 10 atmospheres, after preheating it to 80 deg. cent., and of course it is much cheaper and easier to preheat the air by the exhaust gases, than to subject it to very high compression which cannot be done without the consumption of power.

An interesting peculiarity of this process is that it can be applied to existing motors by changing slightly the injection and valve gear, and thus adapting them to running on cheaper fuels.

There is, however, a limit to the heating of air beyond which it ceases to be of advantage, since, the volume of air admitted to the cylinders being constant, its *weight* will decrease as the temperature rises, and finally a point will be reached when the weight of air admitted will not suffice to insure complete combustion. The article is to be continued.

NEUERE ROHÖLMOTOREN, Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, February 10, 1912. 6½ pp., 16 figs. *d.* Further description of the *latest types of oil engines*. To satisfy the demand for a comparatively *cheap and compact high-speed machine*, the M. A. N. Co. has lately placed on the market a special type built in sizes from 25 to 250 h.p., with two, four, and six cylinders, and piston speed of 3.5 to 4.5 m (11.5 to 14.8 ft.) per second. The largest unit built, a four-cylinder 1000-h.p. engine, has a cylinder diameter of 660 mm (26 in.), and stroke 720 mm (28 in.), and makes 187 r.p.m. All the high-speed engines are made with box-pattern engine beds; the air pumps are placed on the engine bed, and are driven direct from the crankshaft. The pistons of this motor are provided with adjustable blocks, with white metal lining to take care of the play that may be produced by the wear of the piston. The lower part carries the guide block, and the upper all the piston rings, while between the two is a caulking ring permitting the exact regulation of the height of the compression chamber. This arrangement considerably simplifies the manufacture of the pistons since the upper and lower parts may be made in standard sizes, and only the caulking ring has to be individually adjusted for each engine. The upper part of the cylinder is provided with a double deck and oil cooling; the lubrication is by forced feed. Notwithstanding very careful workmanship, the high-speed oil engine has not so far proved an unqualified success. While in operation, the rapidly moving imperfectly balanced machine parts set up powerful vibrations which transmit themselves to the buildings in an undesirable manner. Six-cylinder engines permit perfect balancing, but in four-cylinder engines the foundations must be exceptionally strong, even when the cranks are placed at 90 deg. to

each other, which appears to be better than with cranks arranged symmetrically in the middle of the engine. On the whole, the author thinks that stationary high-speed oil engines are not yet "ripe for the market."

NEUERE ROHÖLMOTOREN, Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, February 24, 1912. 2 pp., 5 figs. *d.* Description of the *Diesel motors* built by Sulzer Brothers in Winterthur, Switzerland. This concern used to build its high-speed Diesel engines with a horizontal compressor, but gave up this practice lately, and now uses a normal three-stage vertical compressor.

GENERATOREN ZUR VERGASUNG VON KOKSLÖSCHE BZW. KOKSGRUS, Meyer. *Journal für Gasbeleuchtung*. January 27, 1912. 7 pp., 12 figs. *d.* Cp. NEUE GASERZEUGER FÜR FEINKÖRNIGE ODER STAUBFORMIGE BRENNSTOFFE, Gwosdz. *Elektrotechnischer Anzeiger*, February 11, 1912. 2 pp., 4 figs. *d.* Cp. The Journal, March 1912, p. 413. Both articles describe gas producers for small coal, mainly on the basis of the practice of the Julius Pintsch Company, the first treating particularly the problem of burning coke waste and coke dross, as well as cinders from locomotive smoke-boxes. The processes and gas producers described have been thoroughly tested in European practice where there are many plants of considerable capacity (over 1000 h.p.) working on this kind of fuel.

Coke waste from gas plants is of fairly large and regular grain, and can be utilized in ordinary shaft producers. Cinders from locomotive smoke-boxes, especially those using good pit coal, are of comparatively large grain, 2 to 4 mm (0.078 to 0.157 in.), while coke dross from coke ovens is generally of very small grain; both have to be burned in specially constructed producers, and can then give 1 h.p-hr. for 0.8 kg (say 1¾ lb.).

The article contains descriptions, in order of historical development, of the types of producers constructed by the Julius Pintsch Company for small grained coal, of which the following are the latest types and deserve particular attention: Fig. 4 shows a clinker-melting producer. The air for combustion is introduced through tuyeres at such a rate that the clinkers melt, and can then be tapped off from below. The main trouble with this type of producers is that small coal, say under 1.5 mm (0.06 in.), is carried away with the stream of gas, due to the high velocity of gases which have to be maintained in the zone of combustion. A better type for use where regularity of work is essential is therefore the step-grate producer (Fig. 5). The fuel must be introduced from above, and the producer provided with a loading device arranged in such a manner that the fuel may be introduced inside without letting the air in. The material is heaped against the step-grate, and by a suitable design of the producer this heaping may be regulated so as to give the most advantageous region of gasification for each material, according to its grain. A zone of incandescence is produced above the grate, the gases pass through the whole depth of fuel layer, and leave the generator at the top. A special system of air regulation allows the opening of the ash pan without affecting the gas production. The gases which come from the producer are led through an evaporator, and there produce steam of very low pressure (1 to 1.5 m, or

40 to 60 in., of water) which passes to a jet blower and drives into the producer as much air as is required for the gasification, thus making the air and steam mixture constant. The advantage of this construction consists in the ability to remove the ashes and clinkers from the grate without in any way interfering with the action of the producer. The size of these producers has to be fairly large, and there is an advantage in building them round. Fig. 6 shows an arrangement for getting rid of the tar

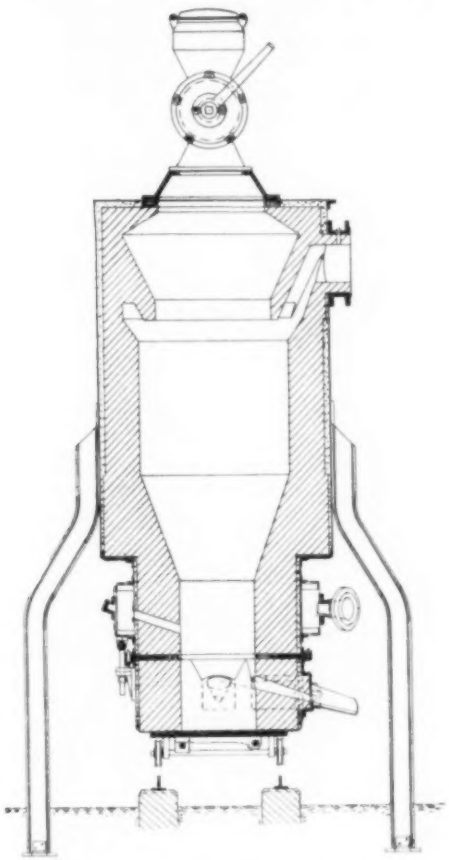


Fig. 4 Pintsch Clinker-Melting Producer

in the fuel. The gas in going from the generator *a* to the tar-separator *b* passes through a scrubber (not shown in the drawing), where it is preheated up to the temperature at which the separation of tar can best take place. The gas then passes to the tar-separator, and is finally driven by an exhauster or ventilator to where it is wanted. There is also a by-pass with a regulator *d* which sends through the tar-separator a full amount of gas even when the load on the engines is suddenly thrown off,

and thus prevents poor separation of tar due to variations in the amount of gas flowing through the separator. The attempt to regulate the action of the separator by drawing the gas bell of the separator in and out did not prove quite successful. The article contains also a description of the

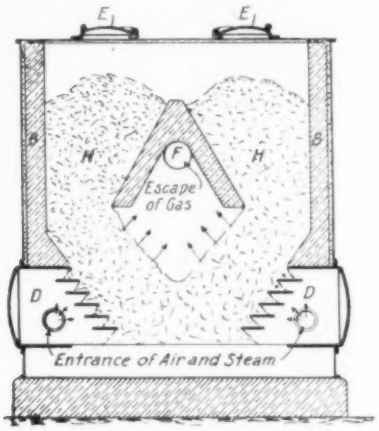


Fig. 5 Pintsch Step-Grate Producer for Small-Grained Coal

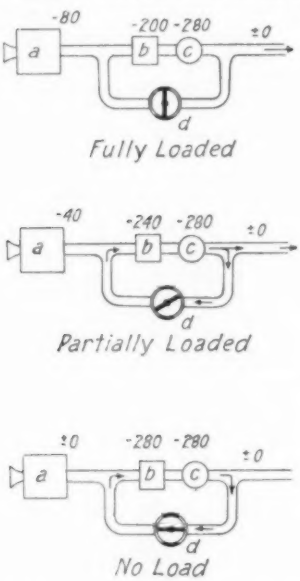


Fig. 6 Tar Separation Diagram

Pintsch suction-gas plant, for which see H. Güldner, Design and Construction of Internal Combustion Engines, New York, 1910, p. 275. The article by Gwosd also contains a short description of the Harnisch inclined-grate producer.

Über Automobil- und Gasmotorenschmieröle, F. Schwartz and H. Schluter. *Zeits. des Mitteleuropäischen Motorwagen-Vereins*, Middle February 1912. 4 pp., 2 of which are tables. *e.* Data from *tests of lubricating oils for automobiles and gas engines* made by the Prussian Royal Laboratory for Testing Materials. There is a considerable difference in the behavior of lubricating oils in steam engines and gas or gasoline motors, due to the very high temperature and great piston speed in the latter case, and, although the processes affecting the lubricating oil in an internal combustion engine are as yet but slightly known, there is no doubt but that part at least of the oil is burned up by the sudden fierce heat of the explosion of the mixture. The combustion of the oil is, however, incomplete, and the more so, the more carbon and the more matter boiling only at a high point, contained in the oil. Most of the oxygen in the air drawn in is required for the combustion of the fuel, and if the oil contains more carbon than the remaining unconsumed oxygen can take care of, the extra carbon is not burned up, but forms a residue, and produces undesirable friction and heating up of parts. The products of incomplete combustion in the exhaust are also, as shown by these experiments, the *cause of its unpleasant smell.*

To test in how far the presence of heavy constituents affected the lubricating properties of oils it was necessary to find a method of separating the light from the heavy oils; for this purpose acetone was used, which has the property to dissolve, at room temperature, heavy oils, but leaves light oils unaffected. In tests made with a Benz automobile, the motor was first made to run idle and with excessive lubrication, and then under regular road conditions. When oil treated with acetone was used as a lubricant, and lubrication excessive, thick gray-white clouds were exhausted, but the smell was very faint and by no means unpleasant, while with a normal amount of lubrication there was no smell at all. When ordinary oil, i. e., oil not treated by acetone, was used in the same manner, the familiar unpleasant smell was immediately noticed. It is therefore quite possible to produce a lubricating oil that would do away with the automobile smell. The article contains complete tables of data from the tests, both for automobile and gas engines.

Stehende Hochleistungs-Diesel-Viertaktmaschine der H. Pauksch A.-G., Landsberg a. W. *Die Gasmotorentechnik*, February 1912. 3 pp., 4 figs. *d.* Description of a *vertical high-speed four-cycle Diesel engine* built by the above named concern. For this type of machine with high piston velocity is predicted a promising future because it can be used both for stationary and marine engines. As compared with the present types it has the advantages of lower initial cost, smaller dimensions, and less weight, the last being particularly important for export trade. Its higher speed is of great advantage where the engine may be directly connected to a dynamo, pump, air compressor, etc., a thing that cannot be done with a slow-speed engine at all. Its disadvantage lies in its higher consumption of fuel as compared with the slow-speed engine, the difference in favor of the latter being about 20 grams (0.44 lb.) per h.p-hr. The vertical construction was selected to obtain as quiet and vibrationless

action as possible, and for the same reason the engine frame and bed-plate were made of cast iron (see Fig. 7).

The high speed of the engine required an installation of a very reliable system of lubrication. The crank bearings have forced-feed lubrication with copper piping, the oil being carried from the crank bearings, through

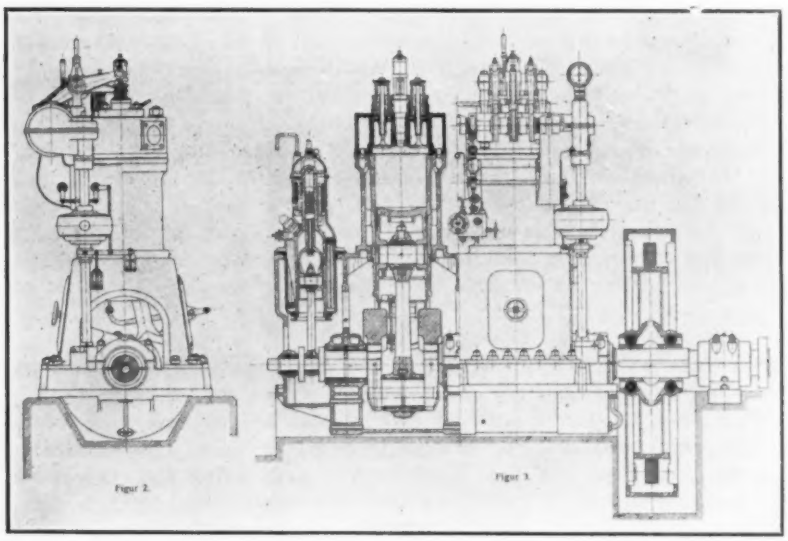


FIG. 7 PAUSKCH VERTICAL HIGH-SPEED FOUR-CYCLE DIESEL ENGINE

ducts in the shaft and connecting rod, to the crank pin and piston pin bearings, the last being also provided with a special lubrication of its own. All other parts which require lubrication receive it from a central lubricating apparatus with an adjustable number of oil-drops. All the oil after use passes into the crank pit, whence it is driven by a rotary pump, through a condenser coil and a filter working at a pressure of three-quarters to one atmosphere, back to the crank bearings: this arrangement reduces the consumption of oil very considerably.

The engine frame is of the usual box pattern, and is cast in one piece with the cylinder jacket. Large openings with door latches provide convenient access to the crank case, the doors on one side forming the point of support of the piston-driven indicator rod. The cylinder liner is protected at the bottom by a cast-in ring of white metal from the action of cooling water that may collect around it. The bottom of the cylinder cover is bell-shaped. This increases its strength and prevents the formation of cracks due to considerable changes in temperature. The discs of the admission and exhaust valves are made very large owing to the great velocity of the air, and therefore are not provided with artificial cooling. Since the valve springs are subjected to very great stresses on account of

rapid and powerful variations of compression and tension, valve stems are equipped with two springs of small diameter wire and large diameter of winding. The packing of the valve needle of the fuel valve and its disc atomizer consists only of vulcanized asbestos rings, while the carefully turned spindle of the admission valve has no stuffing box at all. Owing to lack of room, the admission valve had to be placed in the middle of the cylinder cover; the valve lever therefore lies between the forks of the fuel lever, and is provided with two symmetrically arranged rollers corresponding to the divided cam disc. The admission and exhaust cams are laid out on different curves which insures a very quiet running of the lever rollers, notwithstanding the high speed of the engine.

The condenser coils of the two-stage compressor, both of the high and low-pressure cylinders, are laid in the cooling space of the cylinders in such a way that the stream of hot air from the delivery valves goes direct into the cooled pipes. This gives not only a very effective cooling, but materially simplifies the construction of the compressor. The oil filter, a very important part in a high-speed engine, is said to be efficient and reliable, but no details of its construction are given.

Machine Shop

CLASSIFICATION ET EMPLOI DES MÉTAUX DANS LES ATELIERS DE LA SOCIÉTÉ FRANÇAISE DE LA CONSTRUCTION MÉCANIQUE, J. Netter. *La Technique moderne*, February 1, 1912. 2½ pp., 6 figs. *dp.* The great variety of steel used on the French railroads and the insistence on each particular piece being made of the kind of steel satisfying exactly definite conditions of strength, hardness, etc., necessitated the introduction of a system of classification of materials in large French steel works. This article describes the system used at the Denain works of the *Société française de construction mécanique*. The requirements for such a system are: a method of rapid and exact testing, eliminating as far as possible the personal equation of the operator, and a convenient way of preserving the results so as to have them always ready for reference. After a considerable amount of experimenting it was found that Brinnel's test for hardness is the most simple and reliable, and that, together with a test made by breaking a test piece of standard dimensions by a rotary rammer, it was sufficient for determining the mechanical strength of the material. Three microscopic tests were also made of a section ground off, but not treated; and of metal treated with picric acid and picrate of soda. For each metal four series of tests were made: (*a*) metal in its natural state; (*b*) quenched in water once; (*c*) quenched in water twice; (*d*) quenched in water twice and tempered. The results of the tests are recorded on tablets on which are also placed samples of the metal showing the surface of fracture for each test and the imprint made by the balls in the hardness test, and micrographs from the microscopic tests. A collection of such tablets permits of making, rapidly and with certainty, a selection of the best material available for the given requirements. The article contains descriptions and drawings of some of the testing machines used by that concern.

DIE KÜHLUNG DES WERKZEUGES, N. N. Sawwin. *Dinglers Polytechnisches*

Journal, February 10, 17, 1912. 3 pp., 5 figs. *c.* (Cp. by the same author: ***Zeits. des Oesterr. Ingenieur u. Arch.-Vereines***, No. 32, 1911, ***Revue de Métallurgie***, February 1912, and ***Annals of the Polytechnic Institute of St. Petersburg***, Vol. 3, 1905.) When a steel shaft 300 mm (11.8 in.) thick and 2 m (6.5 ft.) long is turned in a lathe provided with a high-speed steel tool having a linear velocity of 12 m/min. (39 ft. per min.), and cuts a chip 80 qmm (0.1 sq. in.) in cross-section, the pressure at the cutting is 12,000 kg., the useful cutting work 2400 kgm, so that with a 75 per cent efficiency of the lathe the motor must develop over 42 h.p., and the total useful work for one passing of the tool (2 m or 6.5 ft.) is about 5,052,000 kgm (39,790,000 ft-lb.). Nearly all of this work is transformed into heat, and partly dissipated by radiation, partly transmitted to, and absorbed by, the lathe frame, while a large part of it goes to heat the shaving. F. W. Taylor[1] has first shown that the efficiency of a lathe may be increased (up to 40 per cent in cutting steel, and 16 per cent in cutting cast iron) by thoroughly *cooling the cutting tool by a water jet*, but cooling the tool is not all that a jet of liquid has to do. The long shaving of soft steel which stays on the piece for a considerable time exerts against it a strong friction, and that is what the use of the liquid must help to avoid. The exact loss of power due to this friction is as yet unknown, but the above quoted figures of Taylor show that it is very considerable; in the case of cast iron where the chips break off at once the increase of efficiency due to the use of a cooling (and lubricating) liquid was only 16 per cent, while in the case of steel with its long shavings the saving was as high as 40 per cent.

The *liquid used* must satisfy the following *mechanical conditions: (a)* it must efficiently cool the piece turned and cutting tool, and to do this must possess as high a heat conductivity and specific heat as possible; and *(b)* it must act as a good lubricant to take up the immense pressures between the piece turned and the chip, especially in the case of soft metal giving a long shaving. The first condition is best satisfied by water, the second by pure vegetable oil. There are also two *practical conditions: (a)* the liquid must be cheap, and *(b)* it must not attack the metal of the lathe. The author investigated the following liquids: linseed oil; an emulsion of water, oil, soap, and soda; soda solutions; and water. All these liquids were poured in a stream under equal pressure, from above, on to the cutting tool. The largest saving in power (30 per cent as compared with what was required when no lubrication was used) was obtained when linseed oil was used; machine oil came next; other cooling liquids did not appear to diminish the cutting friction to any considerable extent. Saturated soda solution gives the work a good surface, preserves the work and lathe from rusting, and is cheap, and hence makes a very good cooling medium.

It appears therefore that a *cooling and lubricating* agent ought to be used in *cutting steel* (e. g., linseed oil), but *only a cooling* agent in *working cast iron*, and the cheapest of all cooling materials, water, from Professor Sawwin's investigation, appears to be also the most efficient.

[1] On the Art of Cutting Metals, 1906.

PROCÉDÉ POUR RENDRE L'ALUMINUM INATTAQUABLE AUX ACIDES, AUX ALKALIS ET À TOUS AUTRES AGENTS EXTÉRIEURS, M. Pucillo. *La Métallurgie*, February 7, 1912. 1/3 p. *d.* This rather complicated process may be applied to aluminum objects of any form, and consists in the following series of operations: (*a*) the aluminum object is first treated by an alkaline solution, then neutralized by acidulated water, and washed by pure water; (*b*) it is treated by chloride solutions which may be applied in the following order: iron chloride, zinc, tin, and ammonium chloride; (*c*) it is treated by an ammoniacal solution of copper chloride, to which is added some tartrate, such as sodium tartrate or calcium tartrate (5 per cent at least); (*d*) the object is then treated by an alcohol solution of zinc acetate, and neutralized by an aqueous solution of some carbonate, such as potassium carbonate; (*e*) finally it is treated by acidulated water, carefully washed in pure water, and heated to a temperature above 100 deg. cent. (212 deg. fahr.). At this stage the object is improved so far that it may be varnished or electrolytically covered by a metallic deposit. To make it perfectly acid-proof and temperature proof, the following must be done also: (*f*) the object is washed in a mixture of naphtha and drying oil, with some rosin added to it; before being used, this mixture is submitted to the action of oxygen, or of some strong oxidizing agent; (*g*) finally the object is completely dried while hot, and washed with hot caustic soda or another powerful alkaline agent. It is claimed that aluminum which has been so treated will resist practically every outside influence, can be painted by any paint or varnish and chemically cleaned again, etc.

Measurements

MESSUNG HOHER UMLAUFZAHLEN MITTELS DES STROBOSKOPS, Joh. Schillo. *Elektrotechnische Zeits.*, February 15, 1912. *p.* Description of an optical method for measuring very high speeds of rotation. Ordinary tachometers cannot be applied when the speed of rotation exceeds 15,000 r.p.m., as is often the case with steam turbines, and the author recommends the stroboscopic method of measurement, introduced first by Benischke (cp. Electrical Review, London, January 5, 1900) and Wagner. The latter proposed the formula:

$$N.\ z = n.\ Z$$

where N is the number of revolutions per minute of the turbine, z the number of marks on the disc on the turbine-shaft, n the number of revolutions of the motor driving the stroboscop disc, and Z the number of slots on the stroboscop disc. This formula was sufficient as long as the speed of rotation to be measured could also be estimated from other data to a certain extent, but when the speed of rotation became extremely high, the formula ceased to be perfectly reliable, because, as Dr. Wagner himself has shown, a stationary image can be obtained whenever $N:n$ is equal to an integral number. Thus, when $z = 1$, $Z = 20$, and $n = 500$, a stationary image would be obtained when N is equal to 10,000, or 15,000, or 20,000, etc., and in that case there was no way to estimate, by inspection or from the amount of power used, whether the speed was 35,000 or 40,000 r.p.m. In order to make it possible to apply the method to the measurement of very high speeds, the author introduced the auxiliary process of counting the number of images appearing in the stroboscop.

Suppose a black disc with a white mark is placed on the shaft of a motor making 2100 r.p.m., and another disc with a slot in it is placed on the shaft of a motor making from 700 to 2100 r.p.m. (for stroboscopic disc see Fig. 8). If the slot on one disc and white mark on the other are at the beginning of the revolution in the position *I* just opposite each other, and both motors make 2100 r.p.m., they will be again opposite each other in position *II*, or, in other words, the white mark will be visible through the slot twice in each revolution. If the relation of speeds is 2:1, the mark will be seen three times in each revolution; if it is 3:1, one disc passes three parts of the path, while the other makes only one, and the mark will be visible after each quarter of a revolution. This permits

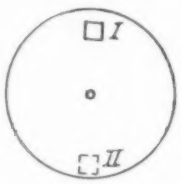


FIG. 8 STROBOSCOPIC DISC

the deduction of the simple formula for the number of times the mark is visible through the slot in the stroboscop disc:

$$A = \frac{N}{n} + 1$$

for rotation in opposite direction and one slot in the disc

$$A = \frac{N}{n} - 1$$

for rotation in the same direction and one slot

$$A = s\left(\frac{N}{n} \pm 1\right)$$

for the case of s slots in the disc or s marks.

This shows that a stationary image may be obtained whenever $s\left(\frac{N}{n} \pm 1\right)$ in an integer. The author gives a table showing what formula for A has to be used for any number of slots and marks, but states that the smaller the number of slots and marks, the simpler the process of measurement and calculation is, the most convenient number both of slots and marks appearing to be 1. For a general description of stroboscops, besides the above quoted article, see Thompson, Light Visible and Invisible, p. 94-99. For a description of a stroboscopic fork and its application to the measurement of speed of rotation of electric machinery see Kennelly, Electrical World, December 26, 1908.

EXAKTE MAAL MED KONSTANT OG PROGRESSIV TOLERANCE, C. F. Helmboe. *Elektroteknisk Tidsskrift*, February 1912. 2 pp., 4 figs. *d.* Description of the *Johansson system of gages based on the principle of progressive tolerance*, or permissible deviation in a standard of measurement, which consists in making the combination gage within the tolerance determined

for that size; this tolerance is further made to correspond with the sum of the tolerances of each of the single gages embodied in the combination gage in question. The coefficient of tolerance is taken as nearly as possible equal to the coefficient of expansion of steel, to avoid error in measurement due to the difference in temperature between the gage and the piece measured. For a general description of the Johansson Universal Combination Gaging Set see American Machinist, October 8, 1908.

Mechanics

EFFETS GYROSCOPIQUES, G. Clauzel. *Revue de mécanique*, January 31, 1912. 20 pp., 28 figs. *mt.A.* The first of a series of articles, being a mathematical and analytical investigation of the so-called *gyroscopic phenomena*. The purpose of the author is to show how the couple of inertia forces in a gyroscope may be determined both as to its direction and as to its dimensions, and from that may be determined the position which the axis of rotation tends to occupy. He goes on to establish the general equations for a figure of revolution rotating about its axis, or a body free to rotate about a point on its supposedly fixed axis. The method applied by the author may be used for the determination of gyroscopic phenomena in the spinning top, the drifting of projectiles, and the rotation of turbines in a vessel or propellers in an airship.

BEITRAG ZUR BERECHNUNG DER SCHRAUBENFEDERN, H. Al. Siebeck. *Zeits. des Vereines deutscher Ingenieure*, December 30, 1911. 4 pp., 5 figs. *t.* A new method is given for determining a spring satisfying given conditions as follows: The efficiency of a spring depends on its greatest safe load and greatest steady deflection. The natural starting point for the computation of a helical spring is the extension diagram (Fig. 9) out of which can be obtained: (*a*) deflection of a single coil in mm.; (*b*) greatest safe load P_e in kg. which approaches more or less the greatest safe steady load in accordance with the variation in the number of deflections per unit time; (*c*) initial load P_v in kg. which may be equal to zero or even negative. $P_e P_v$ represents therefore the actual useful load on the spring, and the coefficient of deflection of the spring h may be expressed as

$$h = \frac{H}{P_v - P_e} \quad \ldots\ldots [1]$$

Therefore h is the compression or extension of one coil of the spring in mm. for change of useful load equal to 1 kg. Practice has shown the extreme importance of correct choice of the coefficient of deflection h, and it is advisable to give the deflection diagram of the springs on drawings of machinery where springs are used, to facilitate their test and installation at the shops.

The calculation of springs is considerably simplified by the introduction of a certain relation between the mean diameter of the coil D and the diameter of the spring wire d. Let this relation be $c = \frac{D}{d}$. The formula for the greatest safe load of the spring

$$P = \frac{\pi}{8} \frac{d^3}{D} K_d \quad \ldots\ldots [2]$$

may then be written in the form

$$P = \frac{\pi}{8} K_d \frac{1}{c} d^2 \quad \ldots\ldots [3]$$

The author shows that by accepting the maximum intensity of stress in wire $K_d = 4070$ kg/qcm (57,835 lb. per sq. in.), he remains always within safe limits; at the same time this allows him to express the formula in the following form, convenient for slide rule calculations:

$$P = 1600 \frac{1}{c} d^2 \quad \ldots\ldots [4]$$

The coefficient of safety S_f of a spring is often determined in practice from

$$S_f = 1 + \frac{n_f}{150} \quad \ldots\ldots [5]$$

where n_f is the number of double deflections per minute (or number per minute of deflections in one direction, i. e. number of compressions or expansions).

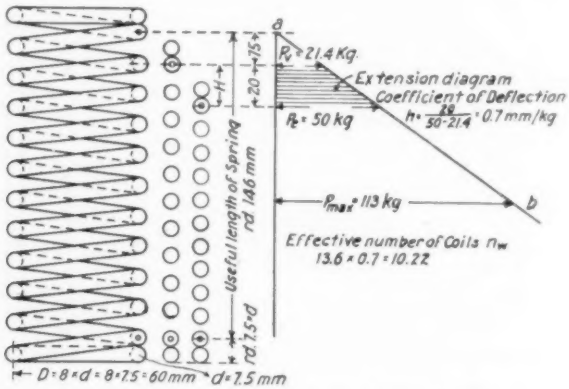


FIG. 9 EXTENSION DIAGRAM OF HELICAL SPRING

For the computation of Table 1 the author accepted certain standard values of $c = \frac{D}{d}$ as shown in the headline of the table, and $h = 1$.

The number of coils required for a spring corresponding to given conditions is obtained from the usual formula:

$$f = \frac{8 n_w D^3 P}{d^4 G} \quad \ldots\ldots [6]$$

where the transverse modulus of elasticity G is taken equal to 800,000 kg/qcm (11,400,000 lb. per sq. in.). By substituting in this formula h for $\frac{f}{P}$, and c for $\frac{D}{d}$, the author again obtains a formula for the number of coils in a spring in a form convenient for slide rule calculations:

$$n_w = \frac{10000}{c^3} dh \quad \ldots\ldots [7]$$

The use of Table 2 can be illustrated by the following example. Re-

quired to find a spring for the exhaust valve of a four-cycle internal-combustion engine, 360 r.p.m. The number of double deflections per minute $n_t = 180$. According to equation [5], the coefficient of safety

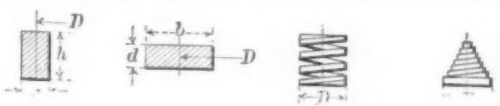
$$S_t = 1 + \frac{n_t}{150} = 1 + \frac{180}{150} = 2.2$$

The deflection diagram is drawn, as in Fig. 1, with a coefficient of deflection $h = 0.7$ mm/kg (0.0125 in. per lb.), in accordance with theoretically required accelerations and practical experience. For a valve lift of 20 mm. (0.78 in.) the load P_e is taken as 50 kg. (110 lb.), and that gives, with a coefficient of safety $S_t = 2.2$, the greatest safe load

$$P_{max} = 2.2\ P_e = 2.2 \times 50 = 110 \text{ kg.} = 242 \text{ lb.}$$

Referring to the upper figures of Table 2, for values approaching that of

TABLE 1 DIAMETER OF WIRE AND NUMBER OF COILS IN CYLINDRICAL AND CONICAL SPRINGS WITH RECTANGULAR CROSS-SECTION OF WIRE

	The Value of the Greatest Safe Load P_{max} of Table 1 must be multiplied by the Following Numbers c for Cylindrical and Conical Springs		The Number of Coils required n'_w with Coefficient of Deflection Unity of Table 1 must be multiplied by the Following Numbers c	
				
	h variable d constant	b variable d constant	For Cylindrical Springs	For Conical Springs
Reference Values of a	1	1	1	4
$a=1$	1.13	1.13	1.41	5.66
$a=1.25$	1.41	1.77	2.16	8.63
$a=1.5$	1.70	2.55	2.94	11.75
$a=1.75$	1.98	3.47	3.73	14.23
$a=2$	2.26	4.53	4.53	18.11
$a=2.5$	2.83	7.07	6.10	24.39
$a=3$	3.40	10.19	7.64	30.56

P_{max} (they are underscored), it is seen that there are many such values, and that they lie practically along a diagonal line. The lower figures show the number of coils required with a coefficient of deflection unity, and since the coefficient of deflection was taken above as $h = 0.7$, the actual number of coils required may vary, according to the Table 2, from 0.7×85.8 to 0.7×0.265, the last figure being, of course, practically out of question. The diameter of the wire may vary from 5 mm. to 17 mm. (0.196 in. to 0.665 in.), and the values $d = 7.5$, $D = 60$ and $n_w = 0.7 \times 14.6 = 10.22$ will probably be chosen as most suitable.

The values in Table 2 are deduced for helical springs with circular cross-section. These values may be applied to springs with square or rectangular cross-section by multiplying them by values in Table 1 corresponding to the nearest value of a.

In Table 2 d is the diameter of the wire, D mean diameter of the coils, the upper figures indicate P_{max} = greatest safe steady load, the lower figures indicate n_w = number of coils required with coefficient of deflection unity.

Steam Engineering

NEUE PATENTE AUF DEM GEBEIT DER DAMPFKESSELFEUERUNG, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, February 2, 16 and 23, 1912. 7 pp., 28 figs. *d*. Brief accounts, with drawings, of new inventions relating to steam boiler firing, patented in Germany during the second half of 1911.

SCHÄDEN AN DAMPFKESSELN, G. Hilliger. *Zeits. für Dampfkessel und Maschinenbetrieb*, February 9, 1912. 2 pp., 4 figs. *cp*. In boilers having a wrought-iron shell and mild steel flues, the flues corroded very rapidly. This was at first ascribed to the use of unsatisfactory material in the flues, but the tests made by the International Association of Boiler Inspection Societies have shown that mild steel alone rusts no more than wrought iron, but that the corrosion is produced by electrolytic action between the two different kinds of materials: an electric potential of 65 millivolts was found between the shell and the material of one of the connecting pieces, this being also the spot most affected by corrosion. It is therefore very important for the life of a boiler to have it made, shell and flues, all of one material. The article also contains a discussion of the losses caused by defective boiler settings, for which see Electrical World, March 2, 1912.

APPAREILS DE MANUTENTION DES CENDRES DANS LES USINES DE L'OUEST-LUMIÈRE, À PUTEAUX (Seine). *Le Génie Civil*, February 3, 1912. 2 pp., 3 figs. *d*. Description of the installation for *handling ashes* at the above-named plant. The furnaces are underground, where the space available is limited, while the output of ashes is about 16 tons per hour. Bucket elevators were at first installed to bring the ashes and clinkers up to the street level direct from the furnace, i. e., hot, but the wear of the parts, due to the combined influence of heat and dust, was so great that this arrangement had to be given up, and electric elevators, partly automatic, partly semi-automatic, installed. These elevators work as follows: The ashes and clinkers are thrown on a grate having square holes, 0.08 m (0.3 in.) on each side, and clinkers larger than that are broken by handbars; from the grate the ashes are conveyed to a loading hopper, and from there to the cage of the elevator. The hopper is provided with a feeding trough oscillating about a horizontal axis; the cage of the elevator in coming down catches this trough and forces it to deliver the ashes into the cage, while a special arrangement prevents the overfilling of the cage. When full, the cage is taken up by an electric winch, and at a certain height is tripped over, and lets the ashes fall into a special reservoir, from which they are later on taken away. Everything works auto-

matically, very little attendance is required, and the installation is said to have given entire satisfaction.

ASSOCIATION ALSACIENNE DES PROPRIÉTAIRES D'APPAREILS À VAPEUR, RAPPORT SUR LES TRAVAUX EXÉCUTÉS EN 1910, V. Kammerer. *Bulletin de la Société Industrielle de Mulhouse*, November 1911. 58 pp., numerous tables. *g. Yearly report of the Alsace Association of Owners of Steam Machinery* containing interesting *data on efficiency of various kinds of mechanical stokers, steaming capacity of boilers, steam consumption of steam engines and turbines of various makes, and heat consumption of gas engines.* Some of the general conclusions formed by the inspecting staff of the Association are of interest. A number of boilers were found to be almost entirely neglected and dirty, and they were located generally in narrow, dark and poorly installed boiler plants. Without going so far as to demand that a boiler plant should be spotless and painted white, there is no doubt that plenty of room, *light and cleanliness* are the very first things to be looked for, because they are not only indispensable for a good upkeep and safety of the plant, but may materially increase its efficiency and economy.

The association is distinctly against the use of *scale solvents* of which the composition is kept secret. Notwithstanding their high sounding names, they are generally made chiefly of soda, and some of them, especially those which contain chlorides, are not only useless, but may do harm. Scale must be dealt with on the basis of an analysis of the water, and not by introducing into the boiler some unknown composition.

After investigating several accidents it was found that it is generally dangerous to lay *steam-carrying pipes* above a certain length horizontally, and that they ought to be laid with considerable inclination towards the drain. It is also safer to make the bends and steam cocks in steel rather than in cast iron.

It is nearly impossible to estimate with any degree of precision the bending stresses to which are submitted *pipe flanges* when, in order to obtain a good joint, the bolts are screwed tightly, especially if the flange has to withstand at the same time stresses due to expansion. As a result, many of the joints in pipes and similar apparatus are working with very low factors of safety, as has been shown lately by the fact that accidents from piping have been far more numerous than from boilers, and with as fatal consequences.

STEHENDE GLEICHSTROM-SCHIEBERDAMPFMASCHINE MIT KURZEM ZYLINDER UND KURZEM KOLBEN, C. Sondermann. *Der praktische Maschinen-Konstrukteur*, February 1, 1912. 1 p., 2 figs. *d.* Description of a *straight-flow steam engine with slide-valve, short cylinder and short piston.* The machine can economically work non-condensing, and is said to give 16 to 20 per cent economy in fuel as compared with reciprocating steam engines with slide valves.

BEITRAGE ZUR BEURTEILUNG DES BETRIEBES VON DAMPFÜBERHITZERN, W. Zimmermann. *Der praktische Maschinen-Konstrukteur*, February 1, 1912. 2 pp., 1 data-sheet. *e.* Account and data of a test on the *efficiency of superheaters*, and conditions impairing it. The two causes found to be the

TABLE 2 DIAMETER OF WIRE AND NUMBER OF COILS IN HELICAL SPRINGS WITH CIRCULAR CROSS-SECTION OF WIRE

$\frac{d}{mm}$	$\frac{D}{d}$ =4	5	6	7	8	9	10	12	14	16	18	20	22.5	25	27.5	30	35	40
	4.0	3.2	2.67	2.29	2.00	1.78	1.60	1.33	1.14	1.00	0.889	0.800	0.711	0.640	0.582	0.533	0.457	0.400
1.0	15.6	8.0	4.63	2.92	1.95	1.37	1.00	0.579	0.364	0.244	0.172	0.125	0.088	0.064	0.048	0.037	0.023	0.016
	9.0	7.2	6.00	5.14	4.50	4.00	3.60	3.00	2.57	2.25	2.00	1.80	1.60	1.44	1.31	1.20	1.03	0.900
1.5	23.4	12.0	6.95	4.37	2.93	2.06	1.50	0.869	0.546	0.366	0.257	0.188	0.132	0.096	0.072	0.056	0.035	0.023
	16.0	12.8	10.7	9.14	8.00	7.11	6.40	5.33	4.57	4.00	3.56	3.20	2.84	2.56	2.33	2.13	1.83	1.60
2.0	31.2	16.0	9.26	5.83	3.90	2.74	2.00	1.16	0.728	0.488	0.343	0.250	0.176	0.128	0.096	0.074	0.047	0.031
	25.0	20.0	16.7	14.3	12.5	11.1	10.0	8.33	7.14	6.25	5.56	5.00	4.44	4.00	3.64	3.33	2.86	2.50
2.5	39.0	20.0	11.6	7.29	4.88	3.43	2.50	1.45	0.910	0.610	0.429	0.313	0.220	0.160	0.120	0.093	0.058	0.039
	36.0	28.8	24.0	20.6	18.0	16.0	14.4	12.0	10.3	9.00	8.00	7.20	6.40	5.76	5.24	4.80	4.11	3.60
3.0	46.8	24.0	13.9	8.75	5.85	4.11	3.00	1.74	1.09	0.732	0.515	0.375	0.263	0.192	0.144	0.111	0.070	0.047
	49.0	39.2	32.7	28.0	24.5	21.8	19.6	16.3	14.0	12.3	10.9	9.80	8.71	7.84	7.13	6.53	5.60	4.90
3.5	54.6	28.0	16.2	10.2	6.83	4.80	3.50	2.03	1.27	0.854	0.600	0.438	0.307	0.224	0.168	0.130	0.082	0.055
	64.0	51.2	42.7	36.6	32.0	28.4	25.6	21.3	18.3	16.0	14.2	12.8	11.4	10.2	9.31	8.53	7.31	6.40
4.0	62.4	32.0	18.5	11.7	7.80	5.48	4.00	2.32	1.46	0.976	0.686	0.500	0.351	0.256	0.192	0.148	0.093	0.062
	81.0	64.8	54.0	46.3	40.5	36.0	32.4	27.0	23.1	20.3	18.0	16.2	14.4	13.0	11.8	10.8	9.26	8.10
4.5	70.2	36.0	20.8	13.1	8.78	6.17	4.50	2.60	1.64	1.10	0.772	0.563	0.395	0.288	0.216	0.167	0.105	0.070
	100	80.0	66.7	57.1	50.0	44.4	40.0	33.3	28.6	25.0	22.2	20.0	17.8	16.0	14.5	13.3	11.4	10.0
5.0	78.0	40.0	23.2	14.6	9.75	6.85	5.00	2.90	1.82	1.22	0.858	0.625	0.439	0.320	0.240	0.185	0.117	0.078
	121	96.8	80.7	69.1	60.5	53.8	48.4	40.3	34.6	30.3	26.9	24.2	21.5	19.4	17.6	16.1	13.8	12.1
5.5	85.8	44.0	25.5	16.0	10.7	7.54	5.50	3.18	2.00	1.34	0.943	0.688	0.473	0.352	0.265	0.204	0.128	0.086
	144	115	96.0	82.3	72.0	64.0	57.6	48.0	41.1	36.0	32.0	28.8	25.6	23.0	20.9	19.2	16.5	14.4
6.0	93.6	48.0	27.8	17.5	11.7	8.22	6.00	3.47	2.18	1.46	1.03	0.750	0.527	0.384	0.289	0.222	0.140	0.094
	169	135	113	96.6	84.5	75.1	67.6	56.3	48.3	42.3	37.6	33.8	30.0	27.0	24.6	22.5	19.3	16.9
6.5	101	52.0	30.1	18.9	12.7	8.91	6.50	3.76	2.37	1.59	1.11	0.813	0.571	0.416	0.313	0.241	0.151	0.101
	196	157	131	112	98.0	87.1	78.4	65.3	56.0	49.0	43.6	39.2	34.8	31.4	28.5	26.1	22.4	19.6
7.0	109	56.0	32.4	20.4	13.7	9.59	7.00	4.05	2.55	1.71	1.20	0.875	0.615	0.448	0.337	0.259	0.163	0.109
	225	180	150	129	113	100	90.0	75.0	64.3	56.3	50.0	45.0	40.0	36.0	32.7	30.0	25.7	22.5
7.5	117	60.0	34.7	21.9	14.6	10.3	7.50	4.34	2.73	1.83	1.29	0.938	0.659	0.480	0.361	0.278	0.175	0.117
	256	205	171	146	128	114	102	85.3	73.1	64.0	56.9	51.2	45.5	41.0	37.2	34.1	29.3	25.6
8.0	125	64.0	37.0	23.3	15.6	11.0	8.00	4.63	2.91	1.95	1.37	1.00	0.702	0.512	0.385	0.296	0.186	0.125
	289	231	193	165	145	128	116	96.3	82.6	72.3	64.2	57.8	51.4	46.2	42.0	38.5	33.0	28.9
8.5	133	68.0	39.4	24.8	16.6	11.6	8.50	4.92	3.09	2.07	1.46	1.06	0.746	0.544	0.409	0.315	0.198	0.133

	324	259	216	185	162	144	130	108	92.6	81.0	72.0	64.8	57.6	51.8	47.1	43.2	37.0	32.4
9.0	140	72.0	41.7	26.2	17.6	12.3	9.00	5.21	3.28	2.20	1.54	1.13	0.790	0.576	0.433	0.333	0.210	0.140
	361	289	241	206	181	160	144	120	103	90.3	80.2	72.2	64.2	57.8	52.5	48.1	41.3	36.1
9.5	148	76.0	44.0	27.7	18.5	13.0	9.50	5.50	3.46	2.32	1.63	1.19	0.834	0.608	0.457	0.352	0.221	0.148
	400	320	267	229	200	178	160	133	114	100	88.9	80.0	71.1	64.0	58.2	53.3	45.7	40.0
10	156	80.0	46.3	29.2	19.5	13.7	10.0	5.79	3.64	2.44	1.72	1.25	0.878	0.640	0.481	0.370	0.233	0.156
	484	387	323	277	242	215	194	161	138	121	108	96.8	86.0	77.4	70.4	64.5	55.3	48.4
11	172	88.0	50.9	32.1	21.5	15.1	11.0	6.37	4.00	2.68	1.89	1.38	0.966	0.704	0.529	0.407	0.256	0.172
	576	461	384	329	288	256	230	192	165	144	128	115	102	92.2	83.8	76.8	65.8	57.6
12	187	96.0	55.6	35.0	23.4	16.4	12.0	6.95	4.37	2.93	2.06	1.50	1.05	0.768	0.577	0.444	0.280	0.187
	676	541	451	386	338	300	270	225	193	169	150	135	120	108	98.3	90.1	77.3	67.6
13	203	104	60.2	37.9	25.4	17.8	13.0	7.53	4.73	3.17	2.23	1.63	1.114	0.832	0.625	0.481	0.303	0.203
	784	627	523	448	392	348	314	261	224	196	174	157	139	125	114	105	89.6	78.4
14	218	112	64.8	40.8	27.3	19.2	14.0	8.11	5.10	3.42	2.40	1.75	1.23	0.896	0.673	0.518	0.326	0.218
	900	720	600	514	450	400	360	300	257	225	200	180	160	144	131	120	103	90.0
15	234	120	69.5	43.7	29.3	20.6	15.0	8.69	5.46	3.66	2.57	1.88	1.32	0.960	0.722	0.555	0.350	0.234
	1020	819	683	585	512	455	410	341	293	256	228	205	182	164	149	137	117	102
16	250	128	74.1	46.6	31.2	21.9	16.0	9.26	5.82	3.90	2.74	2.00	1.40	1.02	0.770	0.592	0.373	0.250
	1160	925	771	661	578	514	462	385	330	289	257	231	206	185	168	154	132	116
17	265	136	78.7	49.6	33.2	23.3	17.0	9.84	6.19	4.15	2.92	2.13	1.49	1.09	0.818	0.629	0.396	0.265
	1300	1040	864	741	648	576	518	432	370	324	288	259	230	207	189	173	148	130
18	281	144	83.3	52.5	35.1	24.7	18.0	10.4	6.55	4.39	3.09	2.25	1.58	1.15	0.866	0.666	0.419	0.281
	1440	1160	963	825	722	642	578	481	413	361	321	289	257	231	210	193	165	144
19	296	152	88.0	55.4	37.1	26.0	19.0	11.0	6.92	4.64	3.26	2.38	1.67	1.22	0.914	0.703	0.443	0.296
	1600	1280	1070	914	800	711	640	533	457	400	356	320	284	256	233	213	183	160
20	312	160	92.6	58.3	39.0	27.4	20.0	11.6	7.28	4.88	3.43	2.50	1.76	1.28	0.962	0.740	0.466	0.312
	1760	1410	1180	1010	882	784	706	588	504	441	392	353	314	282	257	235	202	176
21	328	168	97.2	61.2	41.0	28.8	21.0	12.2	7.64	5.12	3.60	2.63	1.84	1.34	1.01	0.777	0.489	0.328
	1940	1550	1290	1110	968	860	774	645	553	484	430	387	344	310	282	258	221	194
22	343	176	102	64.1	42.9	30.1	22.0	12.7	8.01	5.37	3.77	2.75	1.93	1.41	1.06	0.814	0.513	0.343
	2120	1690	1410	1210	1060	940	846	705	605	529	470	423	376	339	308	282	242	212
23	359	184	107	67.0	44.9	31.5	23.0	13.3	8.37	5.61	3.94	2.88	2.02	1.47	1.11	0.851	0.536	0.359
	2300	1840	1540	1320	1150	1020	922	768	658	576	512	461	410	369	335	307	263	230
24	374	192	111	70.0	46.8	32.9	24.0	13.9	8.74	5.86	4.12	3.00	2.11	1.54	1.15	0.888	0.559	0.374
	2500	2000	1670	1430	1250	1110	1000	833	714	625	556	500	444	400	364	333	286	250
25	390	200	116	72.9	48.8	34.3	25.0	14.5	9.10	6.10	4.29	3.13	2.20	1.60	1.20	0.925	0.583	0.390

most prejudicious to a good working of a superheater were state of impurity of the surface of the superheater tubes and cracks in the superheater brickwork.

EINIGE DAMPFKRAFTANLAGEN MIT ABWÄRMEVERWERTUNG, Max Hottinger. *Zeits. des Vereines deutscher Ingenieure*, January 6, 13 and 20, 1912. 24 pp., 39 figs. *dp.* A series of articles describing the modern German and Swiss *practice in steam power installations with utilization of exhaust steam.* Fig 10 shows a boiler and power plant where exhaust steam is not utilized. From the boilers *K* the steam flows on one side to the high and low-pressure cylinders *H* and *N* of the engine, and from there, through an oil-separator *O*, to the surface or jet condenser K_o, from which the water of condensation flows to the tank *R*. From the other side of the boiler live steam flows into the heating system *Hz*, which may of course be replaced by water preheaters, etc. Before reaching the heating system, the steam passes through the stop-valve *Ab*, steam pressure reducing valve *Rv*, and safety valve *Sv*, the water of condensation being conveyed also into the tank *R*. From there the water passes through the auxiliary tank *Wr*, where as much fresh water is added as is necessary, and is driven by the pump *P*, through the economizer *E*, back into the boiler *K*. Fig. 11 shows the same plant with the difference that it now uses for heating purposes exhaust steam to which low-pressure live steam is added only when the demand for heating exceeds the amount of steam available from the superheat. There is a good oil-separator installed between the receiver and the heating system because oil in heating tubes injures their heat conductivity. No oil-separator is used with steam turbines because their steam is free from oil.

The demand for power may be steady, and that for heat variable, or the reverse, or both may be variable, which necessitates a reliable regulation of the distribution of exhaust steam. The most convenient basis for regulation proved to be the pressure in the exhaust steam main. When the heating system suddenly calls for less steam, or the steam engine wants more to take care of a sudden increase of load, the pressure in the exhaust steam main immediately increases, and this increase of pressure is transferred to the governor *y* (Fig. 12), which automatically increases the admission of steam to the low-pressure cylinder, thus bringing down to the normal level the pressure in the exhaust steam main. With the transfer of part of the load to the low-pressure cylinder the governor *z* maintains the speed of the engine the same as before.

A well designed governor *y* thus becomes the cardinal point of the whole system. The article describes several types which have been built by Sulzer Brothers in Winterthur, Switzerland, and abandoned after trial. The following is the type now in use: A mercury regulator (Fig. 12) in which the steam pressure displaces the mercury, causes rising and sinking of a cast-iron piston *k* in the mercury, weighted with a lead attachment. The piston *k* is connected by lever *a* and rod *b* with the admission valve *c* of the low pressure cylinder, and by rising and sinking in the mercury bath regulates the admission to that cylinder. Any desired pressure in the exhaust steam main, within certain limits, may be obtained

by the movable weight *g*. The steam presses on the mercury and drives it, through the opening *c*, into the pipe *r*, thus changing the degree of immersion of the piston *k*. The device *f* allows to vary the area of the opening *c*, and thus change the resistance offered by it to the passage of the mercury, the purpose of this small opening being to dampen the accidental variations of pressure in the exhaust steam main.

The article contains description of some installations particularly noteworthy for their economic working and full utilization of exhaust steam, and data of tests showing the comparative economy obtained from utilizing exhaust steam. It was found that exhaust steam may best be utilized in

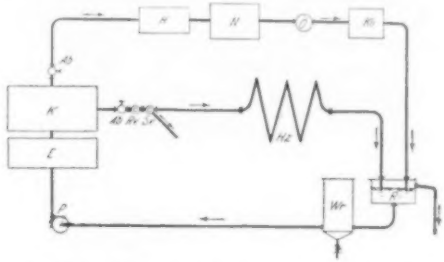


FIG. 10 PLANT WITHOUT UTILIZATION OF EXHAUST STEAM

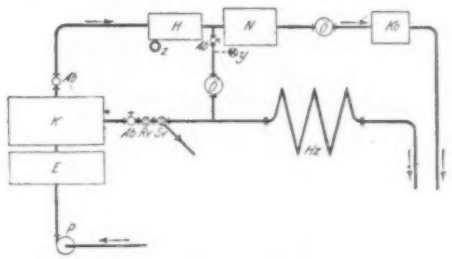


FIG. 11 SAME PLANT WITH UTILIZATION OF EXHAUST STEAM

such plants as breweries, where there is a constant and comparatively regular demand for heat for heating water and for manufacture.

Strength of Materials and Materials of Construction

GRIFFES À SERRAGE AUTOMATIQUE POUR LA MANUTENTION DES TÔLES. *La Métallurgie*, February 7, 1912. 1/3 pp., 2 figs. *d*. Neither hemp ropes nor chains can be conveniently used *for handling sheet-iron plates*, because ropes are easily cut through, and the plates can slip out of the chains. The article describes a convenient and safe *automatic grip* invented for this purpose by Rudolf. It has the form of an ordinary pinch-bar provided with a toothed clamp eccentrically pivoted on one of the jaws of the bar. The iron plate is inserted between the fixed jaw of the bar and the movable clamp and by its weight draws the clamp down until a strong grip is exerted, which makes it practically impossible for the plate to slip out.

Essais sur l'usure des bronzes, Portevin and Nusbaumer. *Revue de métallurgie*, February 1912. 16 pp., 19 figs. *c*. Data and results of tests on the *wearing qualities of various bronzes* rubbing against well lubricated smooth steel. It was found that the wear of ordinary bronzes was proportional to the content of tin; the introduction of phosphorus produces a bronze with better wearing qualities than non-phosphorus bronze with high content of tin, but wearing more than non-phosphorus bronze with low content of tin; the tendency to crystallize by heating is met with in bronze as well as in steel, such crystallization facilitating plastic deformation of the bronze. The article is to be continued.

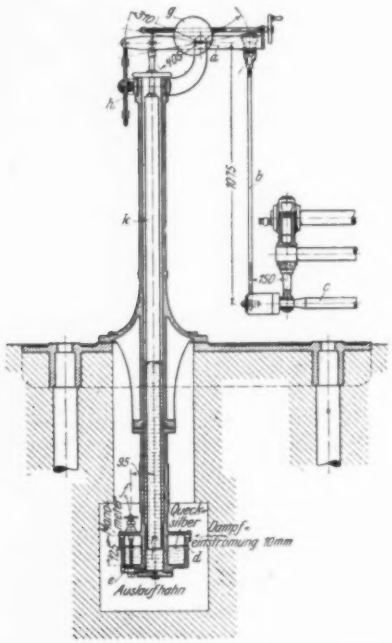


Fig. 12 Mercury Regulator for Pressure in Exhaust Steam Main

Versuche mit umschnürtem Gusseisen, System Dr. v. Emperger. *Beton und Eisen*, February 2, 1912. 3 pp., 21 figs. *c*. Data from tests of "*strapped*" *cast iron*. Neither cast iron nor concrete columns are used much in construction now, the first on account of its brittleness and low resistivity to fire, and the second for the reason that a very slight mistake in construction may lead to the column being weak, and to its breaking down, often with fatal results to the whole structure. Dr. v. Emperger claims to have produced an important new material of construction, which consists in covering a cast-iron column by a concrete jacket strapped all around by a wire helix. In tests which were made by the Austrian Bureau of Reinforced Concrete Construction the cast-iron column broke under a

pressure of 137 tons, or 3420 kg/qcm (say 48000 lb. per sq. in.), while a column of the same size with an external concrete jacket, of 497 qcm (77 sq. in.) total area, broke only under a pressure of 315 tons. The cast-iron reinforcing member may be either a solid column or a tube.

VERLADEBRÜCKE FÜR HOLZ. *Dinglers Polytechnisches Journal*, February 17, 1912. 1½ pp., 2 figs. *d.* Describes a *platform for loading timber into cars.* The traveling motor hoist is a single rail trolley running along an I-shaped rail. The power from the lifting motor to the cable drum is transmitted by a worm gear and two toothed wheels. Besides the load pressure brake in which the braking action is proportional to the load, an electromagnetic brake is installed, the main duty of which is to stop the rotating masses after the motor is switched off, and thus to bring the vertical and side motion of the crane-hooks absolutely under control of the operator. The motors are driven by direct current at 440 volts.

TABELLE ZUR BESTIMMUNG DER SPANNUNGEN IN EISENBETONPLATTEN, E. Suenson. *Beton und Eisen*, February 2, 1912. 1 p. *p.* A table for the determination of *stresses in reinforced concrete slabs.* The author gives a formula which permits, by the use of the table, of finding the stresses both in the iron and the concrete, easily and rapidly. The formulae are convenient for slide-rule calculation.

BALKENVERSUCHE, Dr. F. v. Emperger. *Beton und Eisen*, February 2, 1912. 1½ pp., 3 figs. *t.* The author shows that as yet we know comparatively little as to the *strength of concrete beams* and other structural parts, and attempts to establish a general plan for tests on concrete, and in particular, reinforced concrete beams.

ÜBER DEN EINFLUSS VON ANSTRICHEN AUF DAS ROSTEN DES EISENS, Erik Liebreich and Fritz Spitzer. *Zeits. für Elektrochemie*, February 1, 1912. 5 pp., 8 figs. *e.* The author shows the *fallacy* of the commonly accepted view *that several coats of paint protect iron from rusting better than one.* His experiments with many kinds of paints have uniformly shown that when iron with one coat of paint was as yet practically unaffected by rusting, iron with two coats of paint was partly rusted, with three coats, more so, and with four coats of paint entirely covered by rust. The experiments do not yet permit of offering a well supported explanation of this phenomenon, but the author thinks that the application of several coats of paint, by increasing the number of contacts between the iron and the paint, increases also the number of local electric elements formed at the surface of iron, thus causing large variations in electric potential along the surface, and giving rise to stronger currents than in the case of a single coat of paint. (Cp. Engineering, February 16, 1912, p. 222.)

Testing

UN PROGRAMME D'ESSAIS, M. d'About. *La Technique automobile et aérienne*, February 15, 1912. 3½ pp. *t.* In 1901 The American Society of Mechanical Engineers proposed a *program for testing gas engines.* R. E. Mathot and Ch. de Herbais de Thun, at the Congress at the Liège exhibition, criticized it on the ground that under industrial conditions not all

the tests and measurements recommended by that program can be actually performed. The author of the present article aims to establish a program mainly for the testing of automobile and aeronautical motors, i. e., small two and four-cycle motors having as a rule a high angular velocity. This velocity makes it practically impossible to use indicator diagrams, and the motors must be compared on the basis of effective, and not indicated, power. A complete test, according to the author, must consist of the following operations: (*a*) verification of the organic elements of the machine; (*b*) verification and calibration of the apparatus used; (*c*) measurement of the maximum effective power; (*c*) measurement of the angular velocity corresponding to the maximum effective power; (*d*) measurement of temperatures; (*e*) measurement of consumption. These measurements are in most cases sufficient to determine the quality of a motor, but are not full enough to permit of any general deductions, and without such deductions there can be no progress. The following must therefore be added: (*f*) measurement of power at various speeds of the motor; (*g*) measurement of temperatures for these speeds; (*h*) measurement of consumption of fuel for these speeds; (*i*) determination of the speed requiring minimum consumption of fuel; (*j*) measurement of losses due to friction, with and without compression.

All the results have to be recorded specifically, i. e., on the basis of either a liter of volume of cylinder swept by the piston per revolution, or of a liter of volume of cylinder swept by the piston per minute, or horsepower-hour, or caloric unit of fuel, as the case may be. The measurement of power at various speeds permits the construction of the power curve or characteristic of the motor; the measurement of temperatures and consumption of fuel at various speeds with the last item expressed in specific units of consumption per horsepower-hour and per liter of volume of cylinder swept by the piston gives sufficient data for the construction of a curve of efficiency of the motor; finally, the data on losses by friction, with and without compression, show the mechanical efficiency of the motor and the efficiency of its valve gear. It certainly takes a long time to make all these measurements, but they are neither delicate nor expensive. The article also contains detailed directions for making the measurements required by this program.

Thermodynamics

Über die spezifische Wärme der Gase, Niels Bjerrum. *Zeits. für Elektrochemie.* 3 pp., 1 fig. *t.* This article is a short exposition of a longer work which is promised to appear in the Zeits. für physikalische Chemie, and in which the author proposes, among other things, to show that the *molecular heat of steam* rises with the temperature more than would be expected in accordance with the quanta equation.

Miscellaneous

Les Escaliers Hocquart, A. S. Wadou. *Electro,* January 1912. 4 pp., 7 figs. *d.* Description of the *Hocquart passenger conveyors* for automatically conveying passengers from one floor to another.

Wert des Wassers für Krafterzeugung und für Bewasserung, E.

Krüger. *Zeitschrift des Bayerischen Revisionsvereins*, No. 21, XV Jahrg. *g*. The author shows by a comparative calculation that, *economically, water used for irrigation purposes gives* 10 *to* 55 *times better results than that used for power*, on the supposition that the power installations are large and well laid, which is not always the case, and that irrigated land is used for agricultural purposes. The advantage in favor of irrigated land becomes, however, still more marked when the land is used for higher cultures, production of table vegetables, gardening, etc.

DEUTSCHE MOTORPFLÜGE, K. Praetorius. *Zeits. des Mitteleuropäischen Motorwagen-Vereins*, No. 2, 1912. 3 pp., 7 figs. *d*. Describes a *German motor-plow* invented by Koszoegi, and manufactured by Lanz in Mannheim. It is of the "milling" type, and has a driving shaft with the cutters arranged tangentially to the circle of rotation of the shaft, and working like milling cutters; the cutters not only break the earth, but also push the whole machine forward. This arrangement has the advantage of permitting the machine being built as heavy, and therefore as strong, as desired, and the use of thick cylinder walls and very substantially built machine parts.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XVI)

ARTICLES IN PERIODICALS[1]

ASPHYXIES PAR LES GAZ DES HAUTS-FOURNEAUX, LES, Ad. Breyre. *Annales des Mines de Belgique*, January 1912. 100 pp., 50 figs., 2 tables. *A*.

Asphyxiation caused by the gases from blast furnaces, gas producers, etc. Annual report of accidents, etc., and their causes. Article shows some excellent layouts of gas producer plants, the uses of blast furnace gases to heat boilers, etc.

DIESEL ENGINE TO THE LARGER SEA-GOING VESSELS, THE APPLICATION OF THE, W. Kaemmerer. *Zeitschrift des Vereines deutscher Ingenieure*, January 20, 1912. *chp*.

DIESEL ENGINED OIL BARGE BUILT, FIRST AMERICAN. *International Marine Engineering*, March 1912. 5½ pp., 6 figs.

Account of the 200-ft. steel barge built for the Standard Oil Co.

DIESEL MARINE ENGINE, 100-SHAFT-HORSE-POWER REVERSIBLE. *Engineering*, February 16, 1912. 2 pp., 8 figs.

DIESEL MOTOR SHIP SELANDIA, Axel Holm. *International Marine Engineering*, March 1912. ¾ p., 1 fig.

Describes vessel and engine.

DIESELMOTOREN, THERMODYNAMISCHE UNTERSUCHUNG SCHNELLAUFENDER, Otto Köhler. *Zeitschrift des Vereines deutscher Ingenieure*, February 10, 1912. 2½ pp., 7 curves. *m*.

Thermodynamic investigation of high-speed Diesel engines.

DIESEL UND EXPLOSIONSMASCHINEN BENZISCHER BAUART. *Der praktischer Maschinen Konstructeur*, January 4, 1912. 2 pp., 11 figs. *d*.

Diesel and combustion engines built by Benz. Describes some modern suction gas engines and the Benz crude oil motor. A two-page separate sheet gives very good information about the detail designs of the Benz crude oil motor. See also Foreign Review.

[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

GAS-DRIVEN CARGO VESSEL, SEA-GOING. *The Engineer* (London), February 16, 1912. 2½ pp., 5 figs. *dpA.*

This vessel is 120 ft. long, has a carrying capacity of 350 tons and is equipped with 180-h.p. 6-cylinder vertical gas engines, two 100-h.p. gas producers and necessary auxiliaries. Has been in service for six years, giving 1 i.h.p. per hr. on ¾ lb. bituminous coal.

GASERZEUGERBETRIEB, DIE GEWINNUNG DER NEBENERZEUGNISSE BEIM, Anton Gwiggner. *Stahl und Eisen*, December 21, 1911, and February 1, 1912. *p.*

Article on the recovery of by-products from the operation of gas generators.

GAS GENERATOR PLANT, TESTS OF, K. Neumann. *Zeitschrift des Vereines deutscher Ingenieure*, February 17, 1912. 1½ pp.

Discussion on the above article appearing in *Zeitschrift des Vereines deutscher Ingenieure*, 1911, page 892, by Withol and the author.

KRAFTGASGENERATOREN ANLAGE FÜR FEINKOHLE, H. L. *Braunkohle*, January 12, 1912. 2 pp., 1 fig. *dC.*

Gas producer fed by coal dust. Grate arrangement similar to Roney stoker, allowing comparatively easy passage of the air through the coal dust. See also Foreign Review, March 1912.

LUFTGASHEIZUNG IN DER HAUPTWERKSTATT DELITZSCH, DIE, E. Krause. *Glasers Annalen*, February 1, 1912. 10 pp., 2 figs., 19 tables, 1 curve.

Air-gas heating of the machine shops at Delitzsch.

MARINE GAS ENGINES; THEIR DESIGN AND APPLICATION, E. N. Percy. *International Marine Engineering*, February 1912. 3 pp.

MOTORS FOR LIFEBOATS. *The Engineer* (London), March 1, 1912. 2 pp., 8 figs. 1 table. *dpA.*

Description and particulars of motor-driven lifeboats now in service: the Fay and Brown, Taylor, Thornycroft, Blake and Wolseley engines; with Gaine's, Villinger's and Meissner's reversible propeller; also the Hesse and Savory reverse gear and others.

OIL ENGINE, THE RUMELY. *International Marine Engineering*, March 2, 1912. 2 pp., 2 figs.

Describes the Rumely marine oil engine, 6 cylinder, 125 h.p. rated.

OIL ENGINE, THE TWO-CYCLE SINGLE-ACTING NUREMBERG MARINE. *Engineering*, February 9, 1912. 3⅓ pp., 8 figs.

Describes a 6-cylinder, 150-b.h.p. engine.

PRODUCER, THE MATHOT SUCTION. *Power*, February 27, 1912. 1⅓ pp., 1 fig., 1 table.

A producer which permits of the burning of low-grade fuel of the smallest size. The gas in passing from the top of the producer to a bottom outlet superheats the steam given off by the water vaporizer before reaching the fuel bed, allowing a greater quantity of steam to be dissociated without cooling the gas.

ROHOELMOTOREN, NEUERE, Ch. Pohlmann. *Dinglers Polytechnisches Journal*, February 17 and 24, 1912. 10 pp., 16 figs., 1 table.

New type of crude oil motors. See also Foreign Review, March 1912.

STAUBBESTIMMUNG IM GICHTGAS, UEBER, O. Johannsen. *Stahl und Eisen*, January 4, 1912. 4 pp., 4 figs. *p.*

Article on the determination of dust in blast furnace gases.

TURBINE, A 1000-H.P. GAS. *Engineering News*, February 1, 1912. ¾ p.

General description of internal-combustion turbine built by Brown, Boveri & Co., Mannheim, Germany.

REPORTS OF MEETINGS

ST. LOUIS MEETINGS, MARCH 6 AND 20

The Society coöperated in a meeting conducted by the St. Louis association of members of the American Society of Civil Engineers, held in the Engineers Club on March 6, when The Use of Water Meters was presented by E. E. Wall, assistant water commissioner of St. Louis, and discussed by Edward Flad, Mem.Am.Soc.M.E., consulting engineer, S. Bent Russell, secretary of the Parker-Russell Mining & Manufacturing Company, of St. Louis, Hiram Phillips, consulting engineer, J. R. Cullinane, and others.

On March 20 the Engineers Club conducted a meeting in which the members of the Society participated, Concrete Construction of the East River Wall and Circulating Water Intake at the Ashley Street Station being presented by R. H. Nicholson, construction engineer with the Union Electric Light & Power Company, St. Louis.

PHILADELPHIA MEETING, MARCH 7

A meeting of the Society was held in Philadelphia on March 7th, the American institute of Electrical Engineers and the Electrical Section of the Franklin Institute coöperating. A paper on The Electrification of Main Line Railways was presented by Mr. William J. Clark of New York.

Mr. Clark spoke of the accomplishments of electric traction and the bearing of these on the subject presented, and said that it was difficult to determine just what a main line railroad was. Former steam railroads now electrified are less entitled to qualify as main line roads than many of the so-called electric railways. The paramount reason for electrification will probably be, in the future, the advantageous moving of freight. The total tonnage now hauled by the electric locomotives in operation amounts to over one-third of the entire freight tonnage on the steam roads, although the length of haul is short. In many instances the total cost per ton mile approximates one cent. A distinctive type of the electric main line road is that for towage on canals, the most extensive example of which is the Panama Canal line for towing vessels through the locks. Accuracy and reliability of control secured by these cannot be obtained by mechanical means. There have been great changes in the character of the so-called street railways since 1890, in that these have invaded and expanded the transportation field of main line railways. Making capital investment for electrification has been found cheaper than reducing gradients for steam operation, and cheaper current has been obtained by consolidation of electric railways with power companies. Remarkable results of increased earnings from electrification have been attained on the Manhattan System, and on the Elevated Roads in Chicago, etc. Most of the steam roads now electrified have shown highly satisfactory results.

More electrification has not been accomplished probably because of the diffi-

culty in raising capital and on account of the battle of the systems among electrical engineers. In Europe and Canada where these conditions do not exist much greater progress has been made. The service is one of great public benefit and it is to be regretted that more rapid progress has not been made.

NEW YORK MEETING, MARCH 12

A paper on Practical Problems in Tap and Screw Making by Frank O. Wells, president and treasurer of the Wells Brothers Company, Greenfield, Mass., was presented to H. E. Harris, testing engineer of the company, at a meeting of the Society in New York on March 12. The paper dealt with the importance of size and lead in the cost of assembling manufactured work, as well as the practical limits of accuracy, the reasons for abandoning the V-thread, the proper size of tap drills, the power required for tapping, the effect of lubricants on size and power, and other interesting points.

Among those who discussed the paper were F. G. Coburn; W. R. Porter, Assoc. Am.Soc.M.E., assistant master mechanic of the S. S. White Dental Manufacturing Company, Princes Bay, Staten Island; A. A. Fuller; George W. Adams; Howard Reed; G. A. Gaulison; Horace K. Jones, Mem.Am.Soc.M.E., mechanical superintendent, Corbin Screw Corporation, New Britain, Conn.; Charles B. Russell; J. E. Winter; George B. Pickop, Mem.Am.Soc.M.E., Universal Screw Company, Hartford, Conn.; J. M. Carpenter, Mem.Am.Soc.M.E., president of J. M. Carpenter Tap & Die Company, Pawtucket, R. I.; and written discussions were read from C. B. Buxton, tool supervisor, American Locomotive Company, Providence, R. I.; Wilfred Lewis, Mem.Am.Soc.M.E., president Tabor Manufacturing Company, Philadelphia, Pa.; Luther D. Burlingame, Mem.Am.Soc. M.E., chief draftsman, Brown & Sharpe Company, Providence, R. I.; George W. Adams, Eastman Kodak Company, Rochester, N. Y.

BOSTON MEETING, MARCH 16

At a meeting of the Society in Boston on March 16 a paper, Results of Tests on the Discharge Capacity of Safety Valves, was presented by the author, E. F. Miller, Mem.Am.Soc.M.E., professor of steam engineering of the Massachusetts Institute of Technology, together with an addition to the paper by A. B. Carhart, Mem.Am.Soc.M.E., superintendent of the Crosby Steam Gage & Valve Company of Boston. Charles H. Chase, Mem.Am.Soc.M.E., professor of steam engineering at Tufts College, presented some comparisons of other published tests, and F. H. Keyes, Mem.Am.Soc.M.E., a consulting engineer of Boston, also participated in the discussion.

STUDENT BRANCHES

ARMOUR INSTITUTE OF TECHNOLOGY

On March 6, the Armour Institute of Technology Student Branch was addressed by Sydney V. James, on Scientific Aeroplane Model Testing. The development of the Aeroplane model for experimental purposes was outlined, with examples showing what close approximations have been made to actual flying conditions by this method in the past. A. J. Beerbaum, R. Redlich, A. W. Semerak, J. A. McCague, I. Newman and L. H. Philleo took part in the discussion which followed.

COLUMBIA UNIVERSITY

A paper on The Field for the Mechanical Engineer in Locomotive Manufacture, by L. L. Park, was presented at the February 23 meeting of the Columbia University Student Branch. This was followed by a short talk by Dr. Charles E. Lucke, Mem.Am.Soc.M.E., on the Adaptability of Mechanical Engineering to Non-Mechanical Industries.

CORNELL UNIVERSITY

Wm. T. Magruder, Mem.Am.Soc.M.E., read an interesting paper at the February 19 meeting of Sibley College Student Branch, on Recent Developments in Mechanical Engineering.

At the March 6 meeting, A. R. Nottingham gave a talk, illustrated with lantern slides, on his original investigations in superheated steam. In the discussion which followed, Mr. Nottingham answered points raised by Professor Ellenwood and Messrs. Yoakum, Curtiss and Wegg.

LEHIGH UNIVERSITY

At the February 14 meeting of the Mechanical Engineering Society of Lehigh University, C. K. Baldwin, Mem.Am.Soc.M.E., gave an illustrated lecture on Belt Conveyors.

The following papers were presented at the March 5 meeting: Points of Interest about Anthracite Coal Mining, E. E. Finn (Student 1913); Our Present Measure of Steam Power Plant Economy a Delusion, or the Two Great Fundamental Laws and Boiler Economy, H. J. Williams (Student 1912); Wonders of High Frequency Currents, James Bailey (Student 1912).

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

On February 20, William W. Dodge addressed the Mechanical Engineering Society of the Massachusetts Institute of Technology on How a Man Can Protect an Idea he Wants to Patent. He spoke briefly of the history and then outlined the steps to be taken to patent an idea. He also spoke of the relation of em-

ployer and employee in regard to patentable ideas, and concluded with the legal steps to be taken in cases of infringement and interference.

OHIO STATE UNIVERSITY

The Ohio State University Student Branch held a meeting February 26, at which T. S. Van Voorhis gave a discussion on the Silent Knight Motor, and L. Overturf presented a paper on the Description of and Aerodynamics Applied to the Wright Aeroplane. The interest of the meeting was manifested in the discussions which followed in which Messrs. Burrell, Tom, Stewart, Swarr, Cowser, Spencer, Neilan and Ehrlich took part.

PENNSYLVANIA STATE COLLEGE

Wm. T. Magruder, Mem.Am.Soc.M.E., delivered an address before the Pennsylvania State College Student Branch, March 8, on The Development of Mechanical Engineering in the Year 1911. The members of the civil and electrical engineering departments were invited to attend.

RENSSELAER POLYTECHNIC INSTITUTE

The January meeting of the Rensselaer Polytechnic Institute was merely a short business meeting at which the following officers were elected: W. D. Small, president, R. V. Dodge, vice-president, O. A. Van Denburgh, Jr., secretary, J. C. Hoar, treasurer.

At the February meeting the branch was addressed by A. M. Greene, Jr., Mem.Am.Soc.M.E., who gave a talk on Thermo, giving a new explanation of the term entropy and touching upon the available and unavailable energy in heated gases and vapors.

A special meeting was held on March 13, to which all the mechanical engineering students were invited. The meeting was addressed by W. D. Small, A. M. Greene, Jr., Mem.Am.Soc.M.E., Calvin W. Rice, Secretary Am.Soc.M.E., and R. L. Streeter, Assoc.Am.Soc.M.E. Mr. Rice gave an interesting talk on the value of engineering societies and spoke also of the work of other student branches, and the duties and problems of the graduate engineer. Professor Streeter then gave an illustrated lecture on iron, steel and ore handling machinery and motors, engines, producers, rolls, etc. used in the large steel works.

STATE UNIVERSITY OF KENTUCKY

Frank Daugherty, Jun.Am.Soc.M.E., addressed the State University of Kentucky Student Branch, on the Essentials of Power Plant Design, on February 14. He called attention to the advantages, where large power plants are developed, of the turbine over the reciprocating engine and the considerations involved in the choosing of units.

STEVENS INSTITUTE OF TECHNOLOGY

At a meeting of Stevens Engineering Society held jointly with the Polytechnic Student Section on February 23, William R. Hulbert, Assoc.Am.Soc.M.E., delivered a lecture on Aluminothermics, illustrated with lantern slides and practical demonstrations.

On February 24 the branch inspected the Eastern Division shops of the Brooklyn Rapid Transit Co.

A paper on Metal Working with Punches and Dies was presented by C. R. Barton (Student 1912) at the February 27 meeting. Discussion was offered by E. Byron, O. C. Röesen, C. C. Ross, L. H. Eggert and J. H. Bräutigam.

An inspection trip was made to the Atha Steel Works, Harrison, N. J., on February 28, and on February 29, Col. E. D. Meier, Past-President, Am.Soc.M.E., lectured on Modern Boiler Practice. The speaker was introduced by Dr. D. S. Jacobus, Mem.Am.Soc.M.E.

J. H. Vander Veer (Student 1913), submitted a paper on Electric Car Control at the March 5 meeting. This was followed by discussion by J. Strauss, C. R. Barton, G. L. Clouser, A. E. Bauhan and J. H. Vander Veer.

At the March 14 meeting, W. R. Warner, Past-President, Am.Soc.M.E., gave an address entitled, What are the Astronomers Doing? Mr. Warner was introduced by Dr. John A. Brashear, Hon. Mem.Am.Soc.M.E.

UNIVERSITY OF ILLINOIS

The University of Illinois Student Branch held an annual meeting on February 16, at which the following officers were elected: L. G. Smith, president, A. H. Agard, vice-president, C. A. Schoessel, secretary, M. I. Mix, treasurer. An interesting talk on The Labor Problem in Engineering was given by B. W. Benedict. At this meeting a committee was appointed to draft a new constitution, taking into consideration suggestions made by Calvin W. Rice, Secretary Am.Soc.M.E.

The branch was addressed, March 1, by Prof. O. A. Leutwiler, Mem.Am.Soc. M.E., who gave an illustrated lecture on The Development of Cranes, tracing the growth from the ancient lifting devices to the very large dock-yard cranes of the present day.

At the meeting held March 15, the following papers were presented: The Development of the Internal-Combustion Motor, C. W. Lincoln (Student 1914); Gas Producers, L. M. Mathews (Student 1912); The Automobile Engine, A. Elmendorf (Student 1914). A general discussion followed.

UNIVERSITY OF KANSAS

Papers on the Plant of the Topeka Water Works, E. A. Van Houten and Losses by Condensation in Steam Plants, L. L. Browne, were presented at the February 15 meeting of the University of Kansas Student Branch.

The branch was addressed by Professor Twenhoffel, on March 7, upon the Geology of the Earth's Crust.

At the March 14 meeting the following subjects were presented: The Tool System of the Santa Fé Railroad, M. C. Conley, The Weight per Horsepower of Gas Engines, L. E. Nofsinger, and The Future of the Gas Engine, Mr. Fierabend.

UNIVERSITY OF MISSOURI

The subject discussed at the March 4 meeting of the Student Branch of the University of Missouri was Foundry Practise. It was subdivided into five parts as follows: General Outline, G. D. Mitchell; Pattern Making, H. Mueller; Machine Molding, J. W. Haney; Hand Molding, I. Dunbar; Core Making, F. I.

Kemp. Prof. H. S. Philbrick, Jun.Am.Soc.M.E., and Prof. H. W. Hibbard, Mem.Am.Soc.M.E., followed with discussions, the latter also complimenting the branch on the excellent way in which the subject was handled.

UNIVERSITY OF NEBRASKA

A business meeting of the University of Nebraska Student Branch was held February 22, at which time the following officers were elected: B. R. Beckwith, president, W. H. Noelting, secretary, and P. S. Toney, treasurer. Prof. J. D. Hoffman was appointed honorary chairman to succeed Prof. C. R. Richards.

At the meeting on March 5, the subject discussed was Open-Feed Water Heater versus Closed-Feed Water Heater, E. E. Dale and M. C. Evans arguing for the former and R. J. Foster and Edmond Berger for the latter.

YALE UNIVERSITY

At a meeting of the Yale University Student Branch, held on February 6, Prof. F. R. Hutton, Honorary Secretary Am.Soc.M.E., delivered an illustrated lecture on the subject of The Prevention of Accidents in Factory and Shop.

NECROLOGY

GEORGE WALLACE MELVILLE

Rear Admiral George Wallace Melville, U. S. N., Retired, died at his home in Philadelphia, Pa., March 17, 1912, at the age of seventy-one. An account of his life is published elsewhere in this issue of The Journal.

W. F. BUCK

W. F. Buck, superintendent of motive power of the Atchison, Topeka and Santa Fé Railway, with headquarters at Chicago, Ill., died while traveling through San Bernardino County, California, on January 31, 1912.

Mr. Buck, who was born at St. Andrews, N. B., on June 9, 1863, received his education in the public schools of that community, entering railway service in 1882 as an apprentice in the shops of the Intercolonial Railway, where he received a practical mechanical training. In 1886 he severed his connection with this road and entered the employ of the Northern Pacific Railway as roundhouse foreman at Staples, Minn., and in 1895 he was promoted to the position of general foreman first at Missoula, Mont., and later at Helena, Mont. Mr. Buck steadily advanced in his profession, becoming master mechanic in 1902 of the Rocky Mountain Division of the Northern Pacific Railway, with headquarters at Missoula, and in 1904 of the Arizona division of the Atchison, Topeka & Santa Fé Railway, at Needles, Cal. Several years later he was appointed mechanical superintendent of the Eastern Grand Division of the same company, with headquarters at Topeka, Kansas, and in 1908 entered upon the work in which he was engaged at the time of his death.

Mr. Buck was always deeply interested in his profession and was the designer of a number of practical mechanical devices for railroad work, including a locomotive superheated and a locomotive external throttle.

MILTON P. HIGGINS

Milton P. Higgins, for many years superintendent of the Washburn Shops of Worcester Polytechnic Institute, and widely recognized as the originator of this plan of education as well as for his ability in the industrial and commercial world, died at Worcester, Mass., on March 8, 1912.

Mr. Higgins was born December 7, 1842, in Standish, Me., and early in his boyhood showed his liking for machinery. At the age of seventeen he entered the shops of the Amoskeag Manufacturing Company of Manchester, N. H., and while devoting himself ardently to his work, managed to apply himself to study in the evening and finally to enter and to graduate from Dartmouth College, wholly through his own efforts. After graduation he became a draftsman at the Washburn & Moen Wire Works in Worcester, under the late Charles Hill Morgan, with whom he was associated later for many years at the Worcester Polytechnic Institute.

It was at this time that the plan of Mr. Ichabod Washburn for a shop that would train the students and at the same time be a commercial success was put into practice, and Mr. Higgins was selected by Mr. Morgan and other trustees of the Worcester Polytechnic Institute to carry it out. Industrial educators everywhere and those interested in such matters know how well Mr. Higgins succeeded, building up several lines of business for which he himself designed the machinery, such as machine tools, special grinding apparatus, and the hydraulic elevator of the direct-acting plunger type, and the work of its graduates has been a constant testimony to the thoroughness of administration during the twenty-eight years he was superintendent.

Mr. Higgins' active mind had time for other projects and in 1885 he organized, with George I. Alden, also of Worcester Polytechnic, the Norton Emery Wheel Company, the personnel of which has been altered only by Mr. Higgins' death. The development of the great abrasive industry of the present Norton companies is due largely to the foresight and judgment of Mr. Higgins.

In 1889 Mr. Higgins was granted a year's leave of absence from the Washburn Shops, so that he might organize the shops of the Georgia School of Technology started at that time in Atlanta, Ga., and which is still in successful progress. He was also called upon to plan the shops of the Miller Manual Labor School of Virginia.

In 1896 Mr. Higgins resigned his connection with the Worcester

Polytechnic Institute, although through his appointment later on the Board of Trustees he still maintained his interest in it, and organized the Plunger Elevator Company, which had its inception in the work of the school. Mr. Higgins' genius for organization also showed itself in his formation of the Worcester Pressed Steel Company in 1904.

At the time of his death, in addition to his association with the companies already described, Mr. Higgins was president of the Manchester Supply Company of Manchester, N. H., and of the Sanford Riley Stoker Company of Providence, R. I. He was a director of the Mechanics National Bank of Worcester, and was president of the Board of Trustees of Worcester Polytechnic Institute. He was a charter member of the Society for the Promotion of Industrial Education.

Mr. Higgins was a charter member of this Society and served as one of its vice-presidents from 1901 to 1903. In 1899 he presented before the Society two papers on the Education of Machinists, Foremen and Mechanical Engineers, giving his educational theories, which created wide interest. The Society was represented at Mr. Higgins' funeral by Honorary Vice-Presidents G. I. Rockwood, A. L. Smith, H. B. Smith, C. A. Read, Geo. I. Alden and Paul B. Morgan.

ALBERT W. JACOBI

Albert W. Jacobi was born August 24, 1861, in Newark, N. J. His professional service began in 1878 when he started as an apprentice in the machine trade at the Hewes and Phillips Iron Works of that city. His first two and a half years were spent on the Porter-Allen engines which were at that time built at these works. When Mr. Porter removed to Philadelphia he was placed in charge of the construction of two small engines, and assisted in the erection of numerous machine tools. He then assisted in the drawing room in which department his apprenticeship was completed. During his stay here he realized that his technical education had been much neglected, and therefore decided to attend the Ohio State University. While at college he assisted in designing and constructing the steam heating system in the chemical laboratory as well as in other minor constructions. Upon graduation in 1884 he was reengaged by his former employers, his attention now being confined to machine tools, many of which were of heavy type, to steam engines, boilers and special machinery. He later became identified with the American Lead Pencil Company of Hoboken, as superintendent, but was

obliged to give up this position on account of ill health. In 1901 he became connected with the Gould and Eberhardt Machine Tool Company of Newark, as chief engineer, but three years later relinquished this for a position as superintendent of the Automobile Club of America. In 1908 he entered into private practice as a consulting engineer, in which capacity he was engaged until the time of his death, February 18, 1912.

JOHN G. OULD

John G. Ould died at his home in Brooklyn, N. Y., on December 21, 1911. He was born April 2, 1863, in Falmouth, England, and was educated in the private schools in England and in the Kensington Science and Art School. He also took courses in Pratt Institute and in the Polytechnic Institute of Brooklyn, N. Y. During his youth he was an apprentice in the English shops of A. & W. Robertson, Victoria Docks, London, builders of marine engines; and Waddington and Company, Liverpool. In May 1885 he took the Board of Trade examination for second class engineer, and for the following five years spent most of his time at sea. During this period he was chief engineer of the steamship Montana, sailing between England and Buenos Aires; chief engineer of the steamship Condor, covering the same route; and chief engineer of the steamship Euxine, from England to Constantinople. In June 1890 Mr. Ould came to New York and obtained employment with Seabury and Company and later with the firm of Mitchell and Boyeson as foreman in charge of repairs. He was subsequently engaged as assistant engineer with the Western Union Company and with Ball and Wood Company. In February 1897 he superintended the installation of the mechanical plant of the Polhemus Memorial Clinic and held the position of superintendent and chief engineer of that institution until the time of his death.

EDWARD K. SANCTON

Edward K. Sancton was born in St. John, New Brunswick, June 11, 1840. He was indentured as an apprentice to the Fleming Foundry, machinists and engineers, where he served until 1861. On leaving St. John he entered the employ of the Morgan Iron Works in New York City and was soon placed in charge of the construction and erection of the engines and machinery for the double-turreted monitor, Onondago, and on the completion of this work was made foreman of one of the departments. He later became engineer on

the steamship Fulton, plying between New York and Southampton, this steamer being the pioneer in the attempt to establish a transatlantic line under the American flag subsequent to the Civil War. On the failure of the steamship company profitably to maintain transatlantic service, he returned to shop work and was employed by Fletcher and Harrison, of New York. During this period George W. Quintard, former proprietor of the Morgan Iron Works, sold this establishment and organized a new marine engine works, called the Quintard Iron Works, of which Mr. Sancton was made foreman of the machine department and assistant to the manager. In this position he supervised the construction and erection of many large pumping and marine engines. In the year 1881 he entered the employ of E. D. Leavitt, Jr., consulting and designing engineer, Cambridge, Mass. This engagement was fruitful of an extended experience, the inspection covering work constructing with I. P. Morris Company, in Philadelphia, the Dickson Manufacturing Co., Scranton, Pa., and others, resulting in the reorganization of the Dickson Company in 1883, in the appointment of Mr. Leavitt as consulting engineer, with Mr. Sancton as assistant superintendent and works manager. He continued in this position until 1895 when he was called by Fraser and Chalmers, Limited, of Erith and London, England, to assume the position of general manager of their English works. In 1910 he was appointed a director of this company and took an active interest in their affairs until his death on December 26, 1911.

FRANCIS H. STILLMAN

Francis H. Stillman, president of The Watson-Stillman Company, and a prominent figure in machine tool and engineering industries, died suddenly on February 16, 1912, at his residence in Brooklyn.

Mr. Stillman was born in New York on February 20, 1850, and was graduated from Yale in the class of 1874 with the degree of B.S. On leaving college, Mr. Stillman first associated himself with the Cottrell Printing Press Company, at Westerly, R. I. In 1883 he became president of the firm of Watson & Stillman which succeeded Lyons & Company, and in 1904 the firm was incorporated as The Watson-Stillman Company, Mr. Stillman remaining president up to the time of his death. Under his direction the firm early became prominent in the hydraulic engineering field and has a large and active plant in Aldene, N. J. Mr. Stillman was also president of the Bridgeport Motor Company and of the Pequannock Commercial

Company, and a director of other manufacturing firms. He was one of the recognized pioneers in hydraulic machine tool construction.

Mr. Stillman was a member of the Engineers Club, the Hanover Club of Brooklyn, and the treasurer as well as a director of the National Association of Manufacturers. He organized and was the first president of the Machinery Club of New York, and was also the first president of the National Metal Trades Association.

Mr. Stillman was actively interested in the Gas Power Section of the Society, as well as in the work of its main body, and was serving as the chairman of its executive committee at the time of his death. At a meeting of the committee on February 29, 1912, the following tribute was spread upon the minutes:

> The Committee desires to enter upon its records something of their appreciation of the long and effective services of Mr. Stillman as a member of the Section and of the Executive Committee, and finally its Chairman. He died while carrying the burdens and privileges of that office.
>
> Mr. Stillman brought to the duties of shaping the policy of the Gas Power Section of the Society, his wide experience in administrative affairs, his shrewd and clear-sighted judgment, and that genial kindness of spirit which was one of his greatest charms. In all the difficulties and problems of starting such an enterprise, his abilities and his assiduity of service were splendid factors in bringing the Gas Power Section to its present standard of usefulness and effectiveness. His associates will miss his high standard of business integrity, his keen personal interest in the duties which he had to do, and his willingness to take trouble for the things which he thought worth while. His colleagues will miss also his friendship and the pleasure of intimate association with him.

The Society was represented at Mr. Stillman's funeral by Honorary Vice-Presidents Fred R. Low and Jesse M. Smith.

JOHN BURKITT WEBB

John Burkitt Webb was born in Philadelphia, Pa., November 22, 1841, and entered the Philadelphia high school at the age of thirteen. Although kept at work in a store for some years after leaving school, he was always in pursuit of mechanical and physical problems. His father having patented a windmill, young Webb set about to determine the speed and angle of sails giving the maximum efficiency, and thereby was led to study Hutton's Mathematics, Loomis's Calculus, and other works on higher mathematics. Finally he left the store and went to Bridgeton, N. J., where with Oberlin Smith, he formed a small company to make an electro-magnetic apparatus for playing organs automatically. The enterprise proved too great, however, for the means at hand, and was therefore abandoned. Mr.

Webb then went to work in a pipe-mill and built a trip-hammer which ran many years and greatly expedited a part of the manufacture. At the outbreak of the war business became so dull that he sought employment for a year and a half with a medical journal in Philadelphia, meanwhile studying drawing in the evening at the Franklin Institute. In the summer of 1863, Mr. Webb again started in business at Bridgeton, in partnership with Mr. Smith, building special machine tools. In 1869, he entered the University of Michigan and after graduation was engaged as assistant in the civil engineering department. In 1871, Professor Webb accepted the charge of the Civil Engineering School, newly established in that institution. He held this position for eight years, during which time he made a trip to Europe for the inspection of scientific schools and for the purchase of apparatus, started a small astronomical and meteorological observatory for the school, delivered several courses of original lectures, partially developed his method of treating bridge stresses, and constructed various pieces of apparatus.

In 1879 he resigned his professorship to pursue a course of advanced scientific study abroad, and spent over two years attending lectures on pure mathematics, mathematical physics, logic, etc., and in experimental work in physical laboratories at Heidelberg, Göttingen, Berlin and Paris. A year was spent in Professor Helmholtz's laboratory constructing apparatus for, and working at, the then undeveloped action of an oscillating current through coils of wire and electrolytic solutions, with which considerable progress was made. His skill with tools secured him the privilege of using the apparatus in the instrument-maker's shop of the university, and before leaving Berlin he was made a member of its Mathematical Society. Nearly a year was spent in Paris attending lectures at the Sorbonne and College de France and in examining technical schools, collections and methods there and in other parts of France.

In 1880 Professor Webb was appointed to a new chair of applied mathematics at Cornell University. During his occupancy he delivered original lectures on thermodynamics, mechanism, drawing and drawing instruments, acted as judge at the International Electrical Exhibition of 1884; invented a draught gage which was officially used at this exhibition, and an inertialess steam-engine indicator which excited much attention at the American Association meeting at Montreal, and published an exhaustive article on Belting to Connect Shafts which are not Parallel and do not Intersect.

Professor Webb was called to the position of professor of mathe-

matics and mechanics at Stevens Institute of Technology in 1886. He retired in 1907 under the Carnegie Foundation Pension Fund, and had since that time been engaged in private experimental and consulting engineering work.

In 1888 he originated his floating dynamometer which is a most convenient and perfect device for measuring the power given out or absorbed by motors, dynamos and other machines. In 1892 he invented the viscous dynamometer, which is an absorption dynamometer depending on fluid friction, or viscosity, between rapidly moving surfaces. In 1900 he invented the dynamophone, which is a transmission dynamometer measuring the twist of a shaft carrying power, by a simple telephonic method of great accuracy and reliability.

Professor Webb was a member of the American Association for the Advancement of Science, American Mathematical Society, and the University of Michigan Round Table. He died of pneumonia in New York, February 17, 1912.

FREDERICK W. WOLF

Frederick W. Wolf, who died in Chicago, Ill., February 17, 1912, was born in Duehren, Germany, November 27, 1837. His early education was obtained in the high schools of Weinheim, near Heidelberg, but before being able to enter a technical college he was obliged to take up some specific trade, and in 1852 entered a locksmith shop in Nuremberg, Bavaria, to serve there as an apprentice. Later he went to Switzerland, where he worked in a machine shop, returning to Berlin in order to enter the employ of a sugar machinery manufacturer. At the age of twenty-one he was sent to erect a beet sugar factory 300 miles southwest of Moscow, where he encountered a great many difficulties, being obliged to employ men who could speak only the Russian language. After successfully completing this installation he pursued the work of erecting machinery in different German cities, until he was able to enter the technical college at Carlsruhe in Baden, where he studied mechanical engineering for four years, his last year there being in the capacity of assistant to the professor. After completing his education he was employed by the Guppy Machine Shop, an English firm in Naples, Italy, to design engines for the Italian warships. In 1866 he came to New York and obtained employment as a draftsman in a number of different firms. The next year he went west and for a short period was master mechanic for the Pere Marquette

Railroad Company, at Marquette, Mich. From 1867 to 1875 he was in business as consulting engineer and architect in Chicago. Realizing the immensity of the future in artificial refrigerating machinery, Mr. Wolf purchased in 1875 the right to manufacture the celebrated Linde ice machine, patented by Prof. Carl Linde of the University of Munich. The first Linde machine erected in the United States was imported by Mr. Wolf in October 1882, and installed in the Wacker & Birk Brewery, Chicago, where it still remains. Another machine was imported and erected in the Schlitz Brewery at Milwaukee, but on account of the many difficulties surrounding the importing of these machines, Mr. Wolf decided to begin their manufacture in this country. In 1887 the Fred W. Wolf Company, of which Mr. Wolf was president until his death, was incorporated and continued the successful manufacture of refrigerating machinery.

About 1897, spurred on by the opportunities which presented themselves, and his early training in the erecting of beet sugar factories, the Fred W. Wolf Company, under the direction of Mr. Wolf, installed a number of beet sugar factories in Michigan, which were very successful. On account, however, of the growth of the refrigerating industry, the manufacture of beet sugar machinery was abandoned about 1900, and since that time all the energies of the Fred W. Wolf Company have been devoted to the building and installing of the Wolf-Linde refrigerating and ice-making systems. Over 1200 machines have been built and erected under the direct supervision of Mr. Wolf, and he, without doubt bears the distinction of being the father of the refrigerating industry in the United States.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0151 Member, having engineering sales office handling power equipment and miscellaneous machinery as direct representative of a number of well-known manufacturers, desires to make arrangements, preferably with a young man, to turn over part of line to a suitable person who will have the benefit of all the facilities of the office and is in position to work on a commission basis. Excellent opportunity for a young man capable of building up new business for manufacturers who have not previously been represented in this market. Location, New York.

0152 Three salesmen with engineering experience to cover New Jersey, Ohio and Canadian territory for a prominent manufacturer of ball bearings. Salary depends on man.

0153 Massachusetts concern, manager of shop in manufacture of stand pipes, water tanks and pen stocks. Young engineer, capable of estimating costs of water tanks, towers, etc.

0154 Factory superintendent, for concern manufacturing electrical specialties, and ignition apparatus for gasolene engines. Wants up-to-date man, well educated in mechanical and electrical branches. Location, Massachusetts.

0155 Ohio concern wants high grade men in sales department in application of rubber goods, involving an intimate knowledge of engines, boilers, pumps, transmission machinery, pneumatic tools, etc. Prefers men not over 30 years of age and capable of training to this work.

0156 New York concern wants salesmen of machinery and steam specialties. Men with technical education preferred and two or three years' experience.

0157 Ohio concern desires to secure services of technical graduate who is sufficiently familiar with gas engine work in the smaller sizes to assume successfully the position of general foreman with a view of working up to superintendent. Prefers that applicant have two or three years' practical experience.

0158 Special representative to advance interests in the mechanical rubber goods line, with western mining interests from Mexico to British Columbia,

superior high grade man. One preferred who has had mining and mechanical training making him familiar with plants, people, etc., must be a good "mixer." Will pay salary which will assure a capable man; around age of 30. Will require a great deal of traveling.

0159 Ohio concern desire superintendent with practical shop experience and thorough knowledge of operation of all kinds of tools in both machine shop and foundry; must be capable of handling men; ability in design of jigs, able to reduce costs and increase output.

MEN AVAILABLE

381 Mechanical engineer, age 34, technical graduate, 12 years' extensive experience in design, construction, and operation of steam turbines, condensers, power plants and general machinery. Familiar with electrical work and modern industrial plants. Desires position where executive ability and broad engineering experience can be used to advantage.

382 Professor of experimental engineering, age 33. Eight years' experience in teaching; wide engineering experience. Desires a change to professor of mechanical engineering.

383 Works manager, long experience on light manufacturing involving interchangable parts. Competent to organize all departments of manufacturing plant along modern lines.

384 Sales manager; main experience in boilers, engines and complete power-plant equipment, good general training both technical and business. Wide acquaintance with men worth while in middle and eastern states. Cornell University, M. E.; Mem. Am. Soc. M.E.

385 Young mechanical engineer, technical graduate, shop experience particularly in erecting and testing of hydraulic machinery, desires position with an engineer or with manufacturing concern in erecting, engineering, or sales department.

386 Associate, mechanical, and erecting engineer, technical graduate, age 32, married, three years' experience drafting, two years' erecting plate and wire mills, five years' mechanical engineer of steel car company; at present superintendent of small company. Desires to make a change in the near future.

387 Technical engineer with varied experience, German and American, desires position as assistant to the president of some large enterprise requiring engineering research, or private secretary to man of affairs.

388 Professor of wide experience would be pleased to report on any subject requiring a visit to foreign countries.

389 Factory manager of a concern employing 300 men, would consider making a change. Age 31 years. Graduate Worcester Polytechnic Institute. Familiar with practical cost and production systems. Thorough knowledge of modern grey iron foundry practice. Three years' sales experience.

390 Sales manager, wide experience in boilers, engines and general power-plant equipment lines, desires connection with firm who can appreciate high-grade result producing work. Technical graduate.

391 Graduate U. S. Naval Academy, with experience at sea, later with large concern as supervising engineer in connection with the manufacture of fans, blowers, electric motors, the preparation and issuance of technical literature in connection with same and general charge of sales, desires position as sales manager. Salary $5000.

392 Mechanical engineer, Junior member, German, 29 years old, graduate of the Polytechnicum, Munich. Experience in design and layout of coal and coke handling plants here and abroad. Now connected with one of the largest by-product coke oven concerns in New York City; desires position where experience will be of value or as assistant to the engineer, manager or superintendent in business where there are good chances for advancement.

393 Member, married, graduate with advanced degree prominent eastern technical school; foreign study in technical school, one year. Three years' practical experience. Ten years professor in mechanical engineering department of eastern university, author of text books, desires to make a change. Teaching preferred. Specialties: descriptive geometry, machine design, automobile design and tests.

394 Member, 18 years' experience wants position as superintendent, assistant works manager; age 35, married, practical mechanic, technical education, good hustler and organizer with experience including heaviest class of machinery. At present employed as superintendent.

395 Member, technical graduate, age 34, experience in shipbuilding, steam and gas power machinery, and in teaching these and other mechanical engineering subjects. Prepared to take charge of work connected with technical design and estimates of cost for construction, or problems of economy in existing plants.

396 Member, technical graduate with machine shop training and 15 years' experience in the design, purchasing material and construction of power plants for electric railways, lighting and industrial works, some experience in appraisal work and knowledge of French, German and Swedish; desires position with consulting or construction engineers or manufacturing concern with greater scope for application of experience than present position offers. Would undertake the investigation of plants.

397 Cornell University 1906; Junior member, shop, motor truck, and drafting-room experience. Five years' teaching mechanical engineering. Wishes sales position. Any location.

398 Student member, college graduate in mechanical engineering, experienced in mechanical, structural, architectural and automobile designing and estimating, broad experience in steel mill, experimental engineering, and testing of all kinds of engines, boilers, gas producers, etc.; short while assistant instructor in experimental engineering in college. Desires position with engineering corporation or consulting engineer. Best of references furnished.

399 Graduate engineer, at present manager of plate metal working plant, desires similar position or as assistant manager of larger plant. Broad experience in manufacture of boilers; in engineering and executive departments.

400 Sales engineer, Member, desires to connect with concern requiring mechanical or electrical knowledge combined with the advantage of manufactur-

ing experience that is essential to marketing product and securing its specification with architects, engineers and large manufacturing plants.

401 Chief engineer and designer with 12 years' experience on gas engine and automobile work, technical graduate. Member of Gas Power Section. At present employed, but desires change to larger field of usefulness with either automobile or gas engine manufacturer.

402 Graduate engineer, 38 years old, 15 years' shop and office experience manufacturing iron and steel products, desires position as works manager or superintendent.

403 Junior desires responsible position; age 29, married, at present mechanical engineer in charge of drafting department of large corporation; wide experience in design and construction of laundry, chemical, centrifugal and drying machinery, also fans and blowers; familiar with tests and experiments, cost systems, etc.; able to handle men and to direct them.

404 Junior member desires assistant or full professorship in mechanical engineering; preference for experimental engineering and laboratory work but can cover any kind of mechanical engineering subject. Minimum salary $2500.

405 Engineer of varied experience; cement mill expert; experience in dross mills, hydraulics, electricity, steam, gas and power-house work; reliable, systematic and good organizer.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary Am. Soc. M. E.

ACCIDENT PREVENTION AND RELIEF, F. C. Schwedtman and J. A. Emery. *New York, 1911.*

AMERICAN WATER WORKS ASSOCIATION. Proceedings, 1st–6th, 1881–1886. *Buffalo, 1881–1886.* Gift of Engineering News.

ANNALES DES PONTS ET CHAUSSÉES. Tables générales de la partie technique, 1831–1910. *Paris.*

L'ANNÉE SUCRIÈRE 1909–1910, R. Teyssier. *Paris, 1911.*

APPLIED HEATING AND VENTILATION, WITH CHARTS AND FORMULAS, J. M. Harrison. *New York, 1911.*

ARMOUR ENGINEER. Vol. 3. *Chicago, 1911.*

AUDEL'S ANSWERS ON AUTOMOBILES FOR OWNERS, OPERATORS AND REPAIRMEN, Gideon Harris. *New York, 1912.* Gift of Theo. Audel & Co.

BASES ET MÉTHODES D'ETUDES AEROTECHNIQUES, L. Ventou-Duclaux et M. Robert. *Paris, 1911.*

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Storage Batteries. The Chemistry and Physics of the Lead Accumulator, Harry W. Morse. *New York, 1912.*

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Every librarian or editor of a technical paper will welcome with great joy the issuance of this book. One of the most difficult things that a librarian has to do is to find biographies of living persons, and there has not up to this time existed a biography of living men of science which was international in scope. This book should be particularly interesting to engineers, as for the first time engineers are included among men of science in a biographical dictionary. The biographies of about 100 American engineers are included. In each case there is given, in addition to the ordinary facts found in all biographical dictionaries, the present address of the person, his present position and the publications which he has made, thus forming a valuable bibliography of recent works of science. The work also includes a list of the leading universities of the world, with the names of the professors in all of the important subjects. The price ($2) is remarkably reasonable as compared with works of a similar character.

EXCHANGES

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UNITED ENGINEERING SOCIETY

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The Society wishes to thank Engineering News and the American Electric Railway Association for the large number of interesting books and pamphlets which they have placed at the disposal of the United Engineering Society Library.

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